Gasoline-engine management
Basics and components

BOSCH

Automotive Technology

- EGAS electronic throttle control
- Gasoline direct injection
- NO_x accumulator-type catalytic converter

Imprint

Published by:
© Robert Bosch GmbH, 2001
Postfach 300220,
D-70442 Stuttgart.
Automotive Aftermarket Business Sector,
Department AA/PDI2
Product-marketing, software products,
technical publications.

Editor-in-Chief:
Dipl.-Ing. (FH) Horst Bauer

Editors:
Dipl.-Ing. Karl-Heinz Dietsche,
Dipl.-Ing. (BA) Jürgen Crepin.

Authors:
Dipl.-Ing. Michael Oder
(Basics, gasoline-engine management,
gasoline direct injection),
Dipl.-Ing. Georg Mallebrein (Systems for
cylinder-charge control, variable valve timing),
Dipl.-Ing. Oliver Schlesinger (Exhaust-gas
recirculation),
Dipl.-Ing. Michael Bäuerle (Supercharging),
Dipl.-Ing. (FH) Klaus Joos (Fuel supply,
manifold injection),
Dipl.-Ing. Albert Gerhard (Electric fuel pumps,
pressure regulators, pressure dampers),
Dipl.-Betriebsw. Michael Ziegler (Fuel filters),
Dipl.-Ing. (FH) Eckhard Bodenhausen (Fuel rail),
Dr.-Ing. Dieter Lederer (Evaporative-emissions
control system),
Dipl.-Ing. (FH) Annette Wittke (Injectors),
Dipl.-Ing. (FH) Bernd Kudicke (Types of fuel
injection),
Dipl.-Ing. Walter Gollin (Ignition),
Dipl.-Ing. Eberhard Schnaibel
(Emissions control),
in cooperation with the responsible departments
of Robert Bosch GmbH.

Translation:
Peter Girling.

Unless otherwise stated, the above are all
employees of Robert Bosch GmbH, Stuttgart.

Reproduction, duplication, and translation of this
publication, including excerpts therefrom, is only
to ensue with our previous written consent and
with particulars of source. Illustrations, descrip-
tions, schematic diagrams and other data only
serve for explanatory purposes and for presenta-
tion of the text. They cannot be used as the
basis for design, installation, and scope of deliv-
ery. Robert Bosch GmbH undertakes no liability
for conformity of the contents with national or
local regulations.
All rights reserved.
We reserve the right to make changes.

Printed in Germany.
Imprimé en Allemagne.

1st Edition, September 2001.
English translation of the German edition dated:
February 2001.

Robert Bosch GmbH

Gasoline-engine management
Basics and components

Bosch

Contents

4 Basics of the gasoline (SI) engine
4 Operating concept
7 Torque and output power
8 Engine efficiency

10 Gasoline-engine management
10 Technical requirements
12 Cylinder-charge control
15 A/F-mixture formation
18 Ignition

20 Systems for cylinder-charge control
20 Air-charge control
22 Variable valve timing
25 Exhaust-gas recirculation (EGR)
26 Dynamic supercharging
29 Mechanical supercharging
30 Exhaust-gas turbocharging
33 Intercooling

34 Gasoline fuel injection: An overview
34 External A/F-mixture formation
35 Internal A/F-mixture formation

36 Fuel supply
37 Fuel supply for manifold injection
39 Low-pressure circuit for gasoline direct injection
41 Evaporative-emissions control system
42 Electric fuel pump
44 Fuel filter
45 Rail, fuel-pressure regulator, fuel-pressure damper, fuel tank, fuel lines

48 Manifold fuel injection
49 Operating concept
50 Electromagnetic fuel injectors
52 Types of fuel injection

54 Gasoline direct injection
55 Operating concept
56 Rail, high-pressure pump
58 Pressure-control valve
59 Rail-pressure sensors
60 High-pressure injector
62 Combustion process
63 A/F-mixture formation
64 Operating modes

66 Ignition: An overview
66 Survey
66 Ignition systems development

68 Coil ignition
68 Ignition driver stage
69 Ignition coil
70 High-voltage distribution
71 Spark plugs
72 Electrical connection and interference-suppressor devices
73 Ignition voltage, ignition energy
75 Ignition point

76 Catalytic emissions control
76 Oxidation-type catalytic converter
77 Three-way catalytic converter
80 NO_x accumulator-type catalytic converter
82 Lambda control loop
84 Catalytic-converter heating

85 Index of technical terms
85 Technical terms
87 Abbreviations

Volkswagen Technical Site: http://vwts.ru http://vwts.info

The call for environmentally compatible and economical vehicles, which nevertheless must still satisfy demands for high performance, necessitates immense efforts to develop innovative engine concepts. The increasingly stringent exhaust-gas legislation initially caused the main focus of concentration to be directed at reducing the toxic content of the exhaust gas, and the introduction of the 3-way catalytic converter in the middle of the eighties was a real milestone in this respect.

Just lately though, the demand for more economical vehicles has come to the forefront, and direct-injection gasoline engines promise fuel savings of up to 20%.
This Yellow Jacket technical instruction manual deals with the technical concepts employed in complying with the demands made upon a modern-day engine, and explains their operation.

Another Yellow Jacket manual explains the interplay between these concepts and a modern closed and open-loop control system in the form of the Motronic. This manual is at present in the planning stage.

Basics of the gasoline (SI) engine

The gasoline or spark-ignition (SI) internal-combustion engine uses the Otto cycle[1]) and externally supplied ignition. It burns an air/fuel mixture and in the process converts the chemical energy in the fuel into kinetic energy.

For many years, the carburetor was responsible for providing an A/F mixture in the intake manifold which was then drawn into the cylinder by the downgoing piston.

The breakthrough of gasoline fuel-injection, which permits extremely precise metering of the fuel, was the result of the legislation governing exhaust-gas emission limits. Similar to the carburetor process, with manifold fuel-injection the A/F mixture is formed in the intake manifold.

Even more advantages resulted from the development of gasoline direct injection, in particular with regard to fuel economy and increases in power output. Direct injection injects the fuel directly into the engine cylinder at exactly the right instant in time.

Operating concept

The combustion of the A/F mixture causes the piston (Fig. 1, Pos. 8) to perform a reciprocating movement in the cylinder (9). The name reciprocating-piston engine, or better still reciprocating engine, stems from this principle of functioning.

The conrod (10) converts the piston's reciprocating movement into a crankshaft (11) rotational movement which is maintained by a flywheel (11) at the end of the crankshaft. Crankshaft speed is also referred to as engine speed or engine rpm.

Four-stroke principle

Today, the majority of the internal-combustion engines used as vehicle power plants are of the four-stroke type.

The four-stroke principle employs gas-exchange valves (5 and 6) to control the exhaust-and-refill cycle. These valves open and close the cylinder's intake and exhaust passages, and in the process control the supply of fresh A/F mixture and the forcing out of the burnt exhaust gases.

1st stroke: Induction

Referred to top dead center (TDC), the piston is moving downwards and increases the volume of the combustion chamber (7) so that fresh air (gasoline direct injection) or fresh A/F mixture (manifold injection) is drawn into the combustion chamber past the opened intake valve (5).

The combustion chamber reaches maximum volume (V_h+V_c) at bottom dead center (BDC).

2nd stroke: Compression

The gas-exchange valves are closed, and the piston is moving upwards in the cylinder. In doing so it reduces the combustion-chamber volume and compresses the A/F mixture. On manifold-injection engines the A/F mixture has already entered the combustion chamber at the end of the induction stroke. With a direct-injection engine on the other hand, depending upon the operating mode, the fuel is first injected towards the end of the compression stroke.

At top dead center (TDC) the combustion-chamber volume is at minimum (compression volume V_c).

[1]) Named after Nikolaus Otto (1832-1891) who presented the first gas engine with compression using the 4-stroke principle at the Paris World Fair in 1878.

3rd stroke: Power (or combustion)

Before the piston reaches top dead center (TDC), the spark plug (2) initiates the combustion of the A/F mixture at a given ignition point (ignition angle). This form of ignition is known as externally supplied ignition. The piston has already passed its TDC point before the mixture has combusted completely.

The gas-exchange valves remain closed and the combustion heat increases the pressure in the cylinder to such an extent that the piston is forced downward.

4th stroke: Exhaust

The exhaust valve (6) opens shortly before bottom dead center (BDC). The hot (exhaust) gases are under high pressure and leave the cylinder through the exhaust valve. The remaining exhaust gas is forced out by the upwards-moving piston.

A new operating cycle starts again with the induction stroke after every two revolutions of the crankshaft.

Valve timing

The gas-exchange valves are opened and closed by the cams on the intake and exhaust camshafts (3 and 1 respectively). On engines with only 1 camshaft, a lever mechanism transfers the cam lift to the gas-exchange valves.

The valve timing defines the opening and closing times of the gas-exchange valves. Since it is referred to the crankshaft position, timing is given in "degrees crankshaft". Gas flow and gas-column vibration effects are applied to improve the filling of the combustion chamber with A/F mixture and to remove the exhaust gases. This is the reason for the valve opening and closing times overlapping in a given crankshaft angular-position range.

The camshaft is driven from the crankshaft through a toothed belt (or a chain or gear pair). On 4-stroke engines, a complete working cycle takes two rotations of the crankshaft. In other words, the camshaft only turns at half crankshaft speed.

1 Complete working cycle of the 4-stroke spark-ignition (SI) gasoline engine (example shows a manifold-injection engine with separate intake and exhaust camshafts)

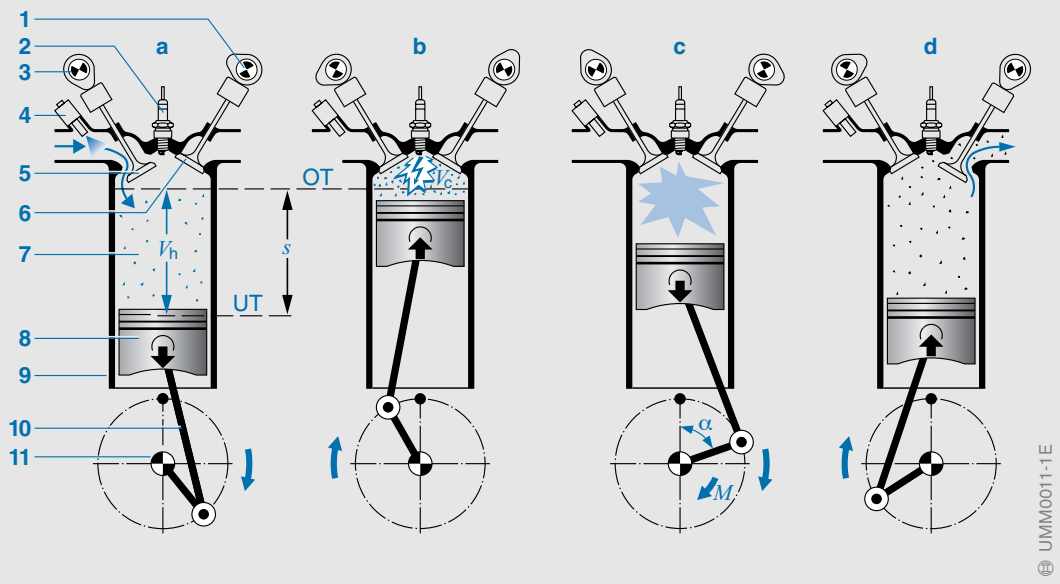


Figure 1

a Induction stroke
b Compression stroke
c Power (combustion) stroke
d Exhaust stroke
1 Exhaust camshaft
2 Spark plug
3 Intake camshaft
4 Injector
5 Intake valve
6 Exhaust valve
7 Combustion chamber
8 Piston
9 Cylinder
10 Conrod
11 Crankshaft
M Torque
α Crankshaft angle
s Piston stroke
V_h Piston displacement
V_c Compression volume

Robert Bosch GmbH

Compression

The compression ratio $\varepsilon = (V_h + V_c)/V_c$ is calculated from the piston displacement V_h and the compression volume V_c.

The engine's compression ratio has a decisive effect upon

- The torque generated by the engine,
- The engine's power output,
- The engine's fuel consumption, and the
- Toxic emissions.

With the gasoline engine, the compression ratio $\varepsilon = 7...13$, depending upon engine type and the fuel-injection principle (manifold injection or direct injection). The compression ratios ($\varepsilon = 14...24$) which are common for the diesel engine cannot be used for the gasoline engine. Gasoline has only very limited antiknock qualities, and the high compression pressure and the resulting high temperatures in the combustion chamber would for this reason cause automatic, uncontrolled ignition of the gasoline. This in turn causes knock which can lead to engine damage.

Air/fuel (A/F) ratio

In order for the A/F mixture to burn completely 14.7 kg air are needed for 1 kg fuel.

2 A/F mixture distribution in the combustion chamber

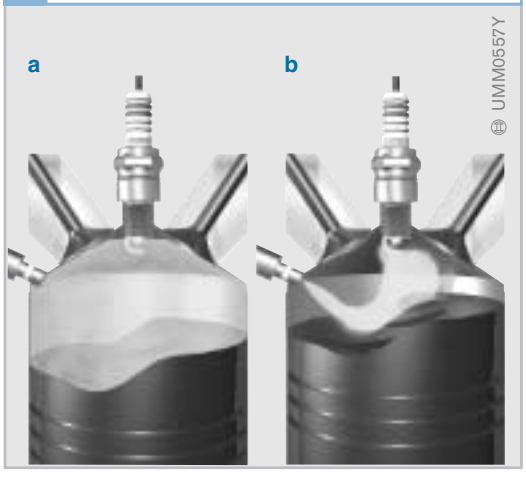


Figure 2
a Homogeneous A/F-mixture distribution
b Stratified charge

This is the so-called stoichiometric ratio (14.7:1).

The excess-air factor (or air ratio) λ has been chosen to indicate how far the actual A/F mixture deviates from the theoretical optimum (14.7:1). $\lambda = 1$ indicates that the engine is running with a stoichiometric (in other words, theoretically optimum) A/F ratio.

Enriching the A/F mixture with more fuel leads to λ values of less than 1, and if the A/F mixture is leaned off (addition of more air) λ is more than 1. Above a given limit ($\lambda > 1.6$) the A/F mixture reaches the so-called lean-burn limit and cannot be ignited.

Distribution of the A/F mixture in the combustion chamber

Homogeneous distribution

On manifold-injection engines, the A/F mixture is distributed homogeneously in the combustion chamber and has the same λ number throughout (Fig. 2a). Lean-burn engines which operate in certain ranges with excess air, also run with homogeneous mixture distribution.

Stratified-charge

At the ignition point, there is an ignitable A/F-mixture cloud (with $\lambda = 1$) in the vicinity of the spark plug. The remainder of the combustion chamber is filled with either a very lean A/F mixture, or with a non-combustible gas containing no gasoline at all. The principle in which an ignitable A/F-mixture cloud only fills part of the combustion chamber is referred to as stratified charge (Fig. 2b). Referred to the combustion chamber as a whole, the A/F mixture is very lean (up to $\lambda \approx 10$). This form of lean-burn operation leads to fuel-consumption savings.

In effect, the stratified-charge principle is only applicable with gasoline direct injection. The stratified charge is the direct result of the fuel being injected directly into the combustion chamber only very shortly before the ignition point.

Torque and output power

Via the cranks on the crankshaft, the conrod converts the piston's reciprocal movement into crankshaft rotational movement. The force with which the expanding A/F mixture forces the piston downwards is converted into torque.

In addition to the force, the lever arm is the decisive quantity for torque. On the internal-combustion engine, the lever arm is defined by the crankshaft throw.

In general, torque is the product of force times lever arm. The lever arm which is effective for the torque is the lever component vertical to the force. Force and lever arm are parallel to each other at TDC, so that the effective lever arm is in fact zero. At a crankshaft angle of 90° after TDC, the lever arm is vertical to the generated force, and the lever arm and with it the torque is at a maximum in this setting. It is therefore necessary to select the ignition angle so that the ignition of the A/F mixture takes place in the crankshaft angle which is characterized by increasing lever arm. This enables the engine to generate the maximum-possible torque.

The engine's design (for instance, piston displacement, combustion-chamber geometry) determines the maximum possible torque M that it can generate. Essentially, the torque is adapted to the requirements of actual driving by adjusting the quality and quantity of the A/F mixture.

The engine's power output P climbs along with increasing torque M and engine speed n. The following applies:

$$P = 2 \cdot \pi \cdot n \cdot M$$

Fig. 1 shows the typical torque and power-output curve, against engine rpm, for a manifold-injection gasoline engine. These diagrams are often referred to in the test reports published in automobile magazines. Along with increasing engine speed, torque increases to its maximum M_{max}. At higher engine speeds, torque drops again. Today, engine development is aimed at achieving maximum torque already at low engine speeds around 2000 min^{-1}, since it is in this engine-speed range that fuel economy is at its highest. Engines with exhaust-gas turbocharging comply with this demand.

Engine power increases along with engine speed until, at the engine's nominal speed n_{nom}, it reaches a maximum with its nominal rating P_{nom}.

The power and torque curves of the internal-combustion (IC) engine make it imperative that some form of gearbox is installed to adapt the engine to the requirements of everyday driving.

1 Example of the power and torque curves of a manifold-injection gasoline engine

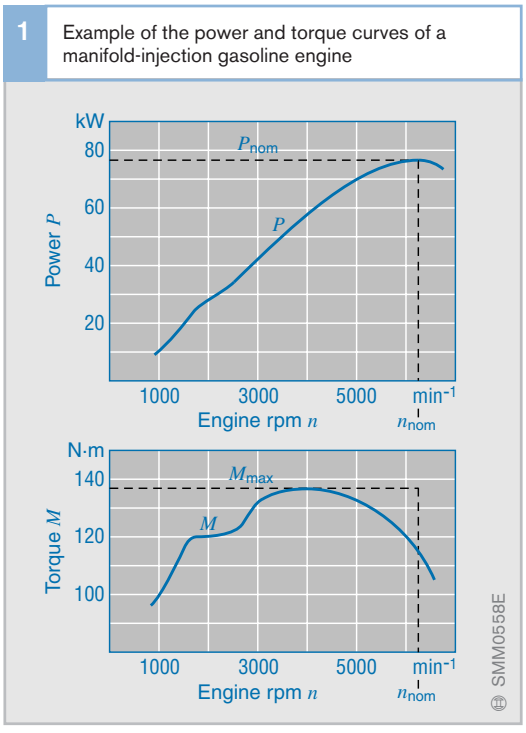


Figure 1
M_{max} Maximum torque
P_{nenn} Nominal power
n_{nenn} Nominal engine speed

Engine efficiency

Thermal efficiency

The internal-combustion does not convert all the energy which is chemically available in the fuel into mechanical work, and some of the added energy is lost. This means that an engine's efficiency is less than 100% (Fig. 1). Thermal efficiency is one of the important links in the engine's efficiency chain.

Pressure-volume diagram (*p-V* diagram)

The *p-V* diagram is used to display the pressure and volume conditions during a complete working cycle of the 4-stroke IC engine.

The ideal constant-volume cycle

Fig. 2 (curve A) shows the compression and power strokes of an ideal process as defined by the laws of Boyle/Mariotte and Gay-Lussac. The piston travels from BDC to TDC (point 1 to point 2), and the A/F mixture is compressed without the addition of heat (Boyle/Mariotte). Subsequently, the mixture burns accompanied by a pressure rise (point 2 to point 3) while volume remains constant (Gay-Lussac).

From TDC (point 3), the piston travels towards BDC (point 4), and the combustion-chamber volume increases. The pressure of the burnt gases drops whereby no heat is released (Boyle/Mariotte). Finally, the burnt mixture cools off again with the volume remaining constant (Gay-Lusac) until the initial status (point 1) is reached again.

The area inside the points 1 – 2 – 3 – 4 shows the work gained during a complete working cycle. The exhaust valve opens at point 4 and the gas, which is still under pressure, escapes from the cylinder. If it were possible for the gas to expand completely by the time point 5 is reached, the area described by 1 – 4 – 5 would represent usable energy. On an exhaust-gas turbocharged engine, the part above the line (1 bar) can to some extent be utilized (1 – 4 – 5′).

Real p-V diagram

Since it is impossible during normal engine operation to maintain the basic conditions for the ideal constant-volume cycle, the actual *p-V* diagram (Fig. 2, curve B) differs from the ideal *p-V* diagram.

Measures for increasing thermal efficiency

The thermal efficiency rises along with increasing A/F-mixture compression. The higher the compression, the higher the pressure in the cylinder at the end of the compression phase, and the larger is the enclosed area in the *p-V* diagram. This area is an indication of the energy generated during the combustion process. When selecting the compression ratio, the fuel's antiknock qualities must be taken into account.

Manifold-injection engines inject the fuel into the intake manifold onto the closed intake valve, where it is stored until drawn into the cylinder. During the formation of the A/F mixture, the fine fuel droplets vaporise. The energy needed for this process is in the form of heat and is taken from the air and the intake-manifold walls. On direct-injection engines the fuel is injected into the combustion chamber, and the energy needed for fuel-droplet vaporization is taken from the air trapped in the cylinder which cools off as a result. This means that the compressed A/F mixture is at a lower temperature than is the case with a manifold-injection engine, so that a higher compression ratio can be chosen.

Thermal losses

The heat generated during combustion heats up the cylinder walls. Part of this thermal energy is radiated and lost. In the case of gasoline direct injection, the stratified-charge A/F mixture cloud is surrounded by a jacket of gases which do not participate in the combustion process. This gas jacket hinders the transfer of heat to the cylinder walls and therefore reduces the thermal losses.

Further losses stem from the incomplete combustion of the fuel which has condensed onto the cylinder walls. Thanks to the insulating effects of the gas jacket, these losses are reduced in stratified-charge operation. Further thermal losses result from the residual heat of the exhaust gases.

Losses at λ =1

The efficiency of the constant-volume cycle climbs along with increasing excess-air factor (λ). Due to the reduced flame-propagation velocity common to lean A/F mixtures, at $\lambda > 1.1$ combustion is increasingly sluggish, a fact which has a negative effect upon the SI engine's efficiency curve. In the final analysis, efficiency is the highest in the range $\lambda = 1.1...1.3$. Efficiency is therefore less for a homogeneous A/F-mixture formation with $\lambda = 1$ than it is for an A/F mixture featuring excess air. When a 3-way catalytic converter is used for efficient emissions control, an A/F mixture with $\lambda = 1$ is absolutely imperative.

Pumping losses

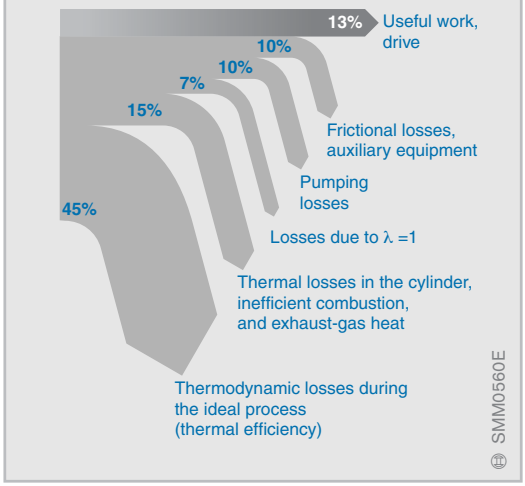
During the exhaust and refill cycle, the engine draws in fresh gas during the 1st (induction) stroke. The desired quantity of gas is controlled by the throttle-valve opening. A vacuum is generated in the intake manifold which opposes engine operation (throttling losses). Since with a gasoline direct-injection engine the throttle valve is wide open at idle and part load, and the torque is determined by the injected fuel mass, the pumping losses (throttling losses) are lower.

In the 4th stroke, work is also involved in forcing the remaining exhaust gases out of the cylinder.

Frictional losses

The frictional losses are the total of all the friction between moving parts in the engine itself and in its auxiliary equipment. For instance, due to the piston-ring friction at the cylinder walls, the bearing friction, and the friction of the alternator drive.

1 Efficiency chain of an SI engine at $\lambda = 1$



2 Sequence of the motive working process in the p-V diagram

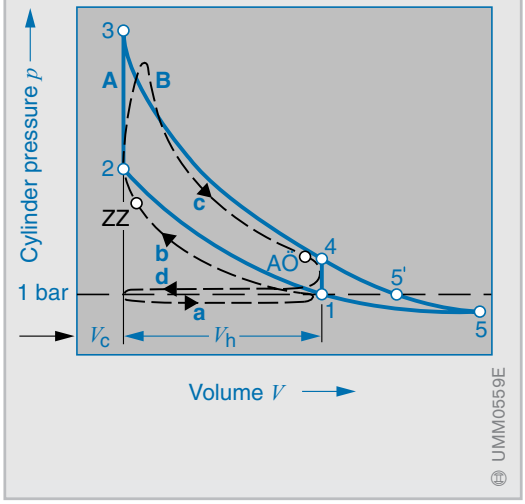


Figure 2
A Ideal constant-volume cycle
B Real p-V diagram
a Induction
b Compression
c Work (combustion)
d Exhaust
ZZ Ignition point
AÖ Exhaust valve opens

Gasoline-engine management

In modern-day vehicles, closed and open-loop electronic control systems are becoming more and more important. Slowly but surely, they have superseded the purely mechanical systems (for instance, the ignition system). Without electronics it would be impossible to comply with the increasingly severe emissions-control legislation.

Technical requirements

One of the major objectives in the development of the automotive engine is to generate as high a power output as possible, while at the same time keeping fuel consumption and exhaust emissions down to a minimum in order to comply with the legal requirements of emissions-control legislation.

Fuel consumption can only be reduced by improving the engine's efficiency. Particularly in the idle and part-load ranges, in which the engine operates the majority of the time, the conventional manifold-injection SI engine is very inefficient. This is the reason for it being so necessary to improve the engine's efficiency at idle and part load without at the same time having a detrimental effect upon the normal engine's favorable efficiency in the upper load ranges. Gasoline direct injection is the solution to this problem.

A further demand made on the engine is that it develops high torque even at very low rotational speeds so that the driver has good acceleration at his disposal. This makes torque the most important quantity in the management of the SI engine.

SI-engine torque

The power P delivered by an SI engine is defined by the available clutch torque M and the engine rpm n. The clutch torque is the torque developed by the combustion process less friction torque (frictional torque in the engine), pumping losses, and the torque needed to drive the auxiliary equipment (Fig. 1).

1 Torque at the drivetrain

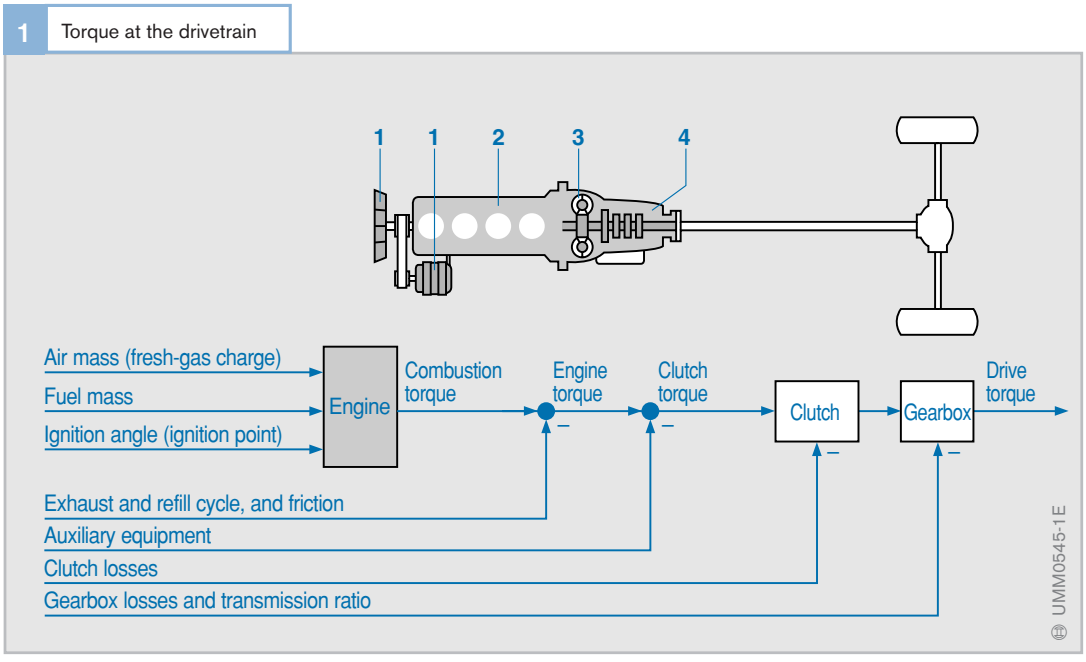


Figure 1
1 Auxiliary equipment (alternator, A/C compressor etc.)
2 Engine
3 Clutch
4 Gearbox

The combustion torque is generated during the power stroke. In manifold-injection engines, which represent the majority of today's engines, it is determined by the following quantities:

- The air mass which is available for combustion when the intake valves close,
- The fuel mass which is available at the same moment, and
- The moment in time when the ignition spark initiates the combustion of the A/F mixture.

The proportion of direct-injection SI engines will increase in the future. These engines run with excess air at certain operating points (lean-burn operation) which means that there is air in the cylinder which has no effect upon the generated torque. Here, it is the fuel mass which has the most effect.

Engine-management assignments

One of the engine management's jobs is to set the torque that is to be generated by the engine. To do so, in the various subsystems (ETC, A/F-mixture formation, ignition) all quantities that influence torque are controlled. It is the objective of this form of control to provide the torque demanded by the driver while at the same time complying with the severe demands regarding exhaust emissions, fuel consumption, power output, comfort and safety. It is impossible to satisfy all these requirements without the use of electronics.

In order that all these stipulations are maintained in long-term operation, the engine management continuously runs through a diagnosis program and indicates to the driver when a fault has been detected. This is one of the most important assignments of the engine management, and it also makes a valuable contribution to simplifying vehicle servicing in the workshop.

Subsystem: Cylinder-charge control

On conventional injection systems, the driver directly controls the throttle-valve opening through the accelerator pedal. In doing so, he/she defines the amount of fresh air drawn in by the engine.

Basically speaking, on engine-management systems with electronic accelerator pedal for cylinder-charge control (also known as EGAS or ETC/Electronic Throttle Control), the driver inputs a torque requirement through the position of the accelerator pedal, for instance when he/she wants to accelerate. Here, the accelerator-pedal sensor measures the pedal's setting, and the "ETC" subsystem uses the sensor signal to define the correct cylinder air charge corresponding to the driver's torque input, and opens the electronically controlled throttle valve accordingly.

Subsystem: A/F-mixture formation

During homogeneous operation and at a defined A/F ratio λ, the appropriate fuel mass for the air charge is calculated by the A/F-mixture subsystem, and from it the appropriate duration of injection and the best injection point. During lean-burn operation, and essentially stratified-charge operation can be classified as such, other conditions apply in the case of gasoline direct injection. Here, the torque-requirement input from the driver determines the injected fuel quantity, and not the air mass drawn in by the engine.

Subsystem: Ignition

The crankshaft angle at which the ignition spark is to ignite the A/F mixture is calculated in the "ignition" subsystem.

Cylinder-charge control

It is the job of the cylinder-charge control to coordinate all the systems that influence the proportion of gas in the cylinder.

Components of the cylinder charge

The gas mixture trapped in the combustion chamber when the intake valve closes is referred to as the cylinder charge. This is comprised of the fresh gas and the residual gas.

The term "relative air charge *rl*" has been introduced in order to have a quantity which is independent of the engine's displacement. It is defined as the ratio of the actual air charge to the air charge under standard conditions (p_0 = 1013 hPa, T_0 = 273 K).

Fresh gas

The freshly introduced gas mixture in the cylinder is comprised of the fresh air drawn in and the fuel entrained with it (Fig. 1). On a manifold-injection engine, all the fuel has already been mixed with the fresh air upstream of the intake valve. On direct-injection systems, on the other hand, the fuel is injected directly into the combustion chamber.

1 Cylinder charge in the gasoline engine

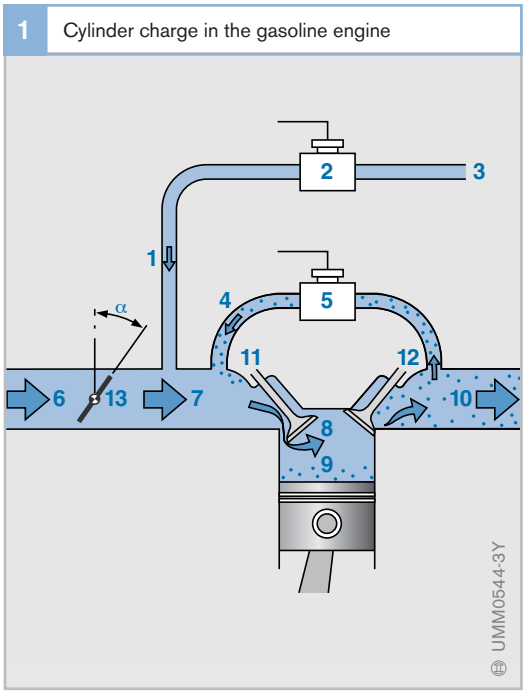


Figure 1
1 Air and fuel vapor (from the evaporative-emissions control system)
2 Canister-purge valve with variable valve-opening cross-section
3 Connection to the evaporative-emissions control system
4 Returned exhaust gas
5 EGR valve with variable valve-opening cross-section
6 Air-mass flow (ambient pressure p_u)
7 Air-mass flow (manifold pressure p_s)
8 Fresh A/F-mixture charge (combustion-chamber pressure p_B)
9 Residual exhaust-gas charge (combustion-chamber pressure p_B)
10 Exhaust gas (exhaust-gas back pressure p_A)
11 Intake valve
12 Exhaust valve
13 Throttle valve
α Throttle valve-angle

The majority of the fresh air enters the cylinder with the air-mass flow (6, 7) via the throttle valve (13) and the intake valve (11). Additional fresh gas, comprising fresh air and fuel vapor, can be directed to the cylinder via the evaporative-emissions control system (3).

For homogeneous operation at $\lambda \leq 1$, the air in the cylinder after the intake valve (11) has closed is the decisive quantity for the work at the piston during the combustion stroke and therefore for the engine's output torque. In this case, the air charge corresponds to the torque and the engine load. During lean-burn operation (stratified charge) though, the torque (engine load) is a direct product of the injected fuel mass.

During lean-burn operation, the air mass can differ for the same torque. Almost always, measures aimed at increasing the engine's maximum torque and maximum output power necessitate an increase in the maximum possible charge. The theoretical maximum charge is defined by the displacement.

Residual gas

The residual-gas share of the cylinder charge comprises that portion of the cylinder charge which has already taken part in the combustion process. In principle, one differentiates between internal and external residual gas. The internal residual gas is that gas which remains in the cylinder's upper clearance volume following combustion, or that gas which is drawn out of the exhaust passage and back into the intake manifold when the intake and exhaust valves open together (that is, during valve overlap).
External residual gas are the exhaust gases which enter the intake manifold through the EGR valve.

Residual exhaust gas comprises inert gas[1]) and, during excess-air operation, unburnt air. The inert gas in the residual exhaust gas does not participate in the combustion during the next power stroke, although it does have an influence on ignition and on the combustion curve.
The selective use of a given share of residual gas can reduce the NO_x emissions.

In order to achieve the demanded torque, the fresh-gas charge displaced by the inert gas must be compensated for by a larger throttle-valve opening. This leads to a reduction in pumping losses which in turn results in a reduction in fuel consumption.

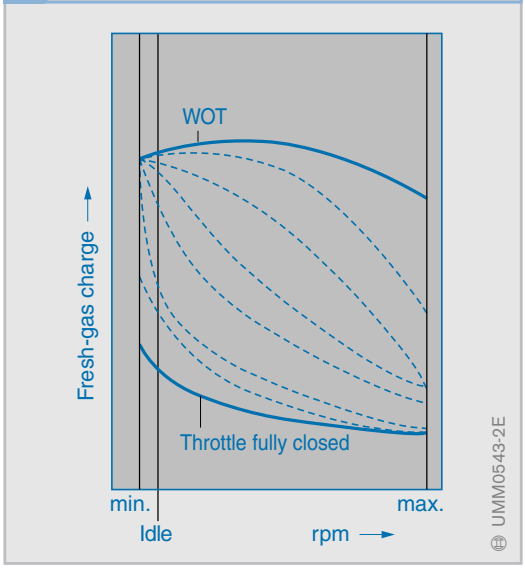
Controlling the fresh-gas charge

Manifold injection

The torque developed by a manifold-injection engine is proportional to the fresh-gas charge. The engine's torque is controlled via the throttle valve which regulates the flow of air drawn in by the engine. With the throttle valve less than fully open, the flow of air drawn in by the engine is throttled and the torque drops as a result. This throttling effect is a function of the throttle valve's setting, in other words its opened cross-section. Maximum torque is developed with the throttle wide open (Wide Open Throttle = WOT).

Fig. 2 shows the principal correlation between fresh-gas charge and engine speed as a function of throttle-valve opening.

2 Throttle characteristic-curve map for an SI engine
– – – Intermediate throttle-valve settings



Direct injection

On direct-injection (DI) gasoline engines during homogeneous operation at $\lambda \leq 1$ (that is, not lean-burn operation), the same conditions apply as with manifold injection.

To reduce the throttling losses, the throttle valve is also opened wide in the part-load range. In the ideal case, there are no throttling losses with the throttle wide open (as it is during full-load operation). In order to limit the torque developed at part load, not all of the air mass entering the cylinder may participate in combustion. In lean-burn applications with excess air ($\lambda > 1$), some of the air drawn in remains as residual exhaust gas in the cylinder or is forced out during the exhaust stroke. In other words, it is not the air charge trapped in the cylinder which is decisive for the developed torque, but rather the fuel injected into the combustion chamber.

[1]) Components in the combustion chamber which behave inertly, that is, do not participate in the combustion process.

Exhaust and refill cycle

The replacement of the used/burnt cylinder charge (= exhaust gas) by a fresh-gas charge takes place using intake and exhaust valves which are opened and closed at precisely defined times by the cams on the camshaft (valve timing). These cams also define the valve-lift characteristic which influences the exhaust and refill cycle and with it the fresh-gas charge which is available for combustion.

Valve overlap, that is, the overlap of the opened times of the intake and exhaust valves, has a decisive influence on the exhaust-gas mass remaining in the cylinder. This exhaust-gas mass also defines the amount of inert gas in the fresh cylinder charge for the next power cycle. In such cases, one refers to "internal" EGR.

The inert-gas mass in the cylinder charge can be increased by "external" EGR. Exhaust pipe and intake manifold are connected by an EGR valve so that the percentage of inert gas in the cylinder charge can be varied as a function of the operating point.

Volumetric efficiency

For the air throughput, the total charge during a complete working cycle is referred to the theoretical charge as defined by the piston displacement. For the volumetric efficiency though, only the exhaust gas actually remaining in the cylinder is considered. Fresh gas drawn in during valve overlap, which is not available for the combustion process, is not considered.

The volumetric efficiency for naturally aspirated engines is 0.6...0.9. It depends upon the combustion-chamber shape, the opened cross-sections of the gas-exchange valves, and the valve timing.

Supercharging

The torque which can be achieved during homogenous operation at $\lambda \leq 1$ is proportional to the fresh gas charge. This means that maximum torque can be increased by compressing the air before it enters the cylinder (supercharging). This leads to an increase in volumetric efficiency to values above 1.

Dynamic supercharging

Supercharging can be achieved simply by taking advantage of the dynamic effects inside the intake manifold. The supercharging level depends on the intake manifold's design and on its operating point (for the most part, on engine speed, but also on cylinder charge). The possibility of changing the intake-manifold geometry while the engine is running (variable intake-manifold geometry) means that dynamic supercharging can be applied across a wide operating range to increase the maximum cylinder charge.

Mechanical supercharging

The intake-air density can be further increased by compressors which are driven mechanically from the engine's crankshaft. The compressed air is forced through the intake manifold and into the engine's cylinders.

Exhaust-gas turbocharging

In contrast to the mechanical supercharger, the exhaust-gas turbocharger is driven by an exhaust-gas turbine located in the exhaust-gas flow, and not by the engine's crankshaft. This enables recovery of some of the energy in the exhaust gas.

Robert Bosch GmbH

A/F-mixture formation

The A/F-mixture formation system is responsible for calculating the fuel mass appropriate to the amount of air drawn into the engine. This fuel is metered to the engine's cylinders through the fuel injectors.

A/F mixture

To run efficiently, the gasoline engine needs a given air/fuel (A/F) ratio. Ideal, theoretically complete combustion takes place at a mass ratio of 14.7:1, which is also referred to as the stoichiometric ratio. In other words, 14.7 kg of air are needed to burn 1 kg of fuel. Or, expressed in volumes, approx. 9,500 liters of air are needed to completely burn 1 liter of gasoline.

Excess-air factor λ

The excess-air factor λ has been chosen to indicate how far the actual A/F-mixture deviates from the theoretically ideal mass ratio (14.7:1). λ defines the ratio of the actually supplied air mass to the theoretical air mass required for complete (stoichiometric) combustion.

$\lambda = 1$: The inducted air mass corresponds to the theoretically required air mass.

$\lambda < 1$: This indicates air deficiency and therefore a rich A/F mixture. On a cold engine, it is necessary to enrich the A/F mixture by adding fuel to compensate for the fuel that has condensed on the cold manifold walls (manifold-injection engines) and cold cylinder walls and which, as a result, is not available for combustion.

$\lambda > 1$: This indicates excess air and therefore a lean A/F mixture. The maximum value for λ that can be achieved is defined by the so-called lean-misfire limit (LML), and is highly dependent upon the engine's design and construction, as well as upon the mixture-formation system used. At the lean-misfire limit the A/F mixture is no longer combustible, and this marks the point at which misfire starts. The engine begins to run very unevenly, fuel consumption increases dramatically, and power output drops.

Other combustion conditions prevail on direct-injection (DI) engines, and these are thus able to run with considerably higher λ figures.

Operating modes

Homogeneous ($\lambda \leq 1$): On manifold-injection engines, the A/F mixture in the manifold is drawn in past the open intake valve during the induction stroke. This leads to an essentially homogeneous mixture distribution in the combustion chamber.

This operating mode is also possible with DI gasoline engines, the fuel being injected into the combustion chamber during the induction stroke.

Homogeneous lean ($\lambda > 1$): The A/F mixture is distributed homogeneously in the combustion chamber with a defined level of excess air.

Stratified charge: This operating mode and those given below are only possible with direct-injection gasoline engines. Fuel is injected only shortly before the ignition point, and an A/F-mixture cloud forms in the vicinity of the spark plug.

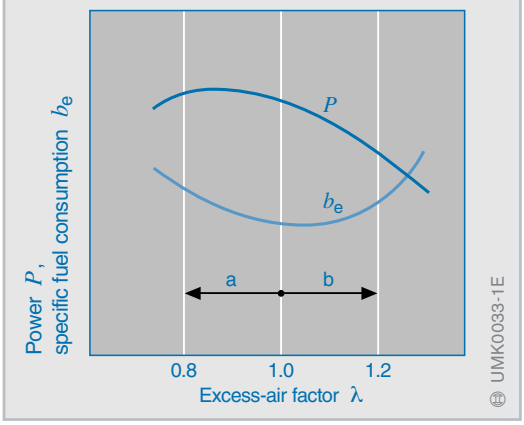
Homogenous stratified charge: In addition to the stratified charge, there is a homogeneous lean A/F mixture throughout the complete combustion chamber. Dual injection is applied to achieve this form of A/F-mixture distribution.

Homogeneous anti-knock: Here, dual injection is also used to achieve an A/F-mixture distribution which to a great extent prevents combustion knock.

Stratified-charge/catalyst heating: Retarded (late) injection leads to the rapid warm-up of the catalytic converter.

Robert Bosch GmbH

1 Influence of the excess-air factor λ on the power P and on the specific fuel consumption b_e under conditions of homogeneous A/F-mixture distribution



2 Effect of the excess-air factor λ on the pollutant composition of untreated exhaust gas under conditions of homogeneous A/F-mixture distribution

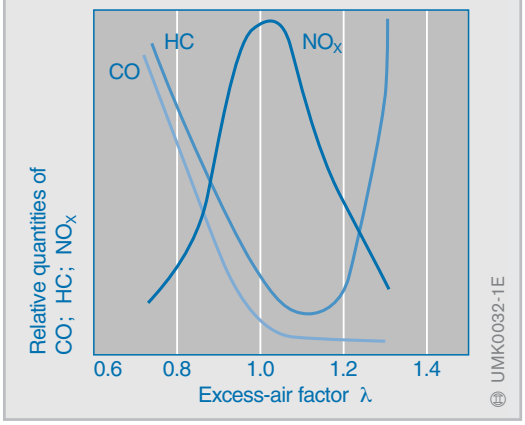


Figure 1
a Rich A/F mixture (air deficiency)
b Lean A/F mixture (excess air)

Specific fuel consumption, power and exhaust emissions

Manifold injection

Manifold-injection gasoline engines develop their maximum power output at 5...15 % air deficiency ($\lambda = 0.95...0.85$), and their lowest fuel consumption at 10...20 % excess air ($\lambda = 1.1...1.2$). Figs. 1 and 2 indicate the extent to which power output, fuel consumption, and exhaust emissions are all a function of the excess-air factor λ. It is immediately apparent that there is no excess-air factor at which all factors are at their "optimum". Best-possible fuel consumption together with best-possible power output are achieved with excess-air factors of $\lambda = 0.9...1.1$.

When a 3-way catalytic converter is used for the treatment of the exhaust gases, it is absolutely imperative that $\lambda = 1$ is maintained precisely when the engine has warmed-up. In order to comply with these requirements, the mass of the intake air must be measured exactly and a precisely metered fuel quantity injected.

An optimal combustion process though not only demands precision fuel injection, but also a homogeneous A/F mixture, which in turn necessitates efficient atomization of the fuel. If the fuel is not perfectly atomized, large fuel droplets are deposited on the walls of the manifold and/or combustion chamber. Since these fuel droplets cannot burn completely, they lead to increased HC emissions.

Gasoline direct injection

For gasoline direct injection, during homogeneous operation at $\lambda \leq 1$, the same conditions apply as with manifold injection. With stratified-charge operation though, a practically stoichiometric A/F mixture is only present in the stratified-charge mixture cloud near the spark plug. Outside this area, the combustion chamber is filled with fresh air and inert gas. Regarding the combustion chamber as a whole, the A/F mixture ratio is very high ($\lambda > 1$).

Since the complete combustion chamber is not filled with a combustible A/F mixture in this operating mode, torque output and power output both drop. Similar to manifold injection, maximum power can only be developed when the complete combustion chamber is filled with a homogeneous A/F mixture.

Depending upon the combustion process, and the A/F-mixture distribution in the combustion chamber, NO_x emissions are generated in the lean-burn mode which cannot be reduced by the 3-way catalytic converter. Here, for emissions control, it is

necessary to take additional measures which call for a NO_x accumulator-type catalytic converter.

Engine operating modes

In some engine operating modes, the fuel requirement differs considerably from the steady-state requirements with the engine at operating temperature. This makes it necessary to take corrective measures in the A/F-mixture formation.

Start and warm-up

When starting with the engine cold, the inducted A/F-mixture leans-off. This is the due not only to inadequate mixing of the intake air with the fuel, but also to the fuel having less tendency to evaporate at low temperatures, and the pronounced wall wetting (condensation of the fuel) on the still-cold intake manifold (only on manifold-injection engines) and on the cylinder walls. To compensate for these negative effects, and to facilitate engine start, additional fuel must be provided during the cranking process.

Even after the engine has started, additional fuel must continue to be injected until it reaches operating temperature. This also applies to the gasoline direct-injection engine. Depending upon the engine's design and the combustion process, stratified-charge lean-burn operation is only possible with the engine at operating temperature.

Idle and part load

Once they have reached their operating temperature, conventional manifold-injection engines all run on a stoichiometric A/F mixture at idle and part load. On direct-injection gasoline engines though, the objective is to run the engine as often as possible with a stratified-charge. This is feasible at idle and at part load, the two operating modes with the highest potential for saving fuel, where fuel savings of as much as 40 % can be achieved with lean-burn operation.

Full load

Essentially, the conditions for manifold injection and gasoline direct injection are pretty much the same at full load. At WOT, it may be necessary to enrich the A/F mixture. As can be seen from Fig. 1, this permits the generation of maximum-possible torque and power.

Acceleration and deceleration

With manifold injection, the fuel's tendency to evaporate depends to a large extent upon the manifold pressure. This leads to the development of a fuel film (wall film) on the intake manifold in the vicinity of the intake valves. Rapid changes in manifold pressure, as occur when the throttle-valve opening changes suddenly, lead to changes in this wall film. Heavy acceleration causes the intake-manifold pressure to increase so that the fuel's evaporation tendency deteriorates, and the wall film thickens as a result. Being as a portion of the fuel has been deposited to form the wall film, the A/F mixture leans-off temporarily until the wall film has stabilized. Similarly, sudden deceleration leans to enrichment of the A/F mixture since the drop in manifold pressure causes a reduction in the wall film and the fuel from the wall film is drawn into the cylinder. A temperature-dependent correction function (transitional compensation) is used to correct the A/F mixture so as to ensure not only the best possible driveability, but also the constant A/F ratio as needed for the catalytic converter.

Wall-film effects are also encountered at the cylinder walls. With the engine at operating temperature though, they can be ignored on direct-injection gasoline engines.

Overrun

At overrun (trailing throttle), the fuel supply is interrupted (overrun fuel cutoff). Apart from saving fuel on downhill gradients, this protects the catalytic converter against overheating which could result from inefficient and incomplete combustion.

Robert Bosch GmbH

Ignition

It is the job of the ignition to ignite the compressed A/F-mixture at exactly the right moment in time and thus initiate its combustion.

Ignition system

In the gasoline (SI) engine, the A/F mixture is ignited by a spark between the electrodes of the spark plug. The inductive-type ignition systems used predominantly on gasoline engines store the electrical energy needed for the ignition spark in the ignition coil. This energy determines how long (dwell angle) the current must flow through the ignition coil to recharge it. The interruption of the coil current at a defined crankshaft angle (ignition angle) leads to the ignition spark and the A/F-mixture combustion.

In today's ignition systems, the processes behind the ignition of the A/F mixture are electronically controlled.

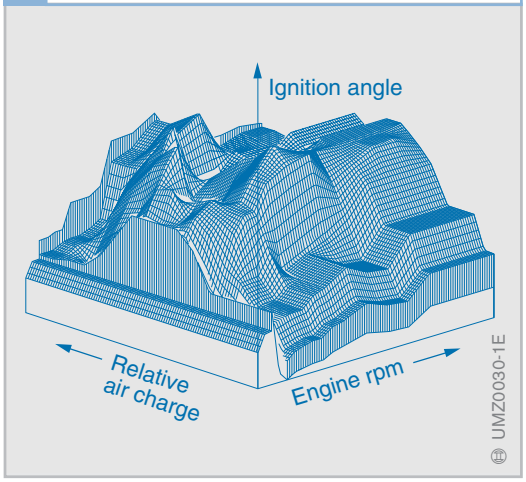
Ignition point

Changing the ignition point (ignition timing)

Following ignition, about 2 milliseconds are needed for the A/F mixture to burn completely. The ignition point must be selected so that main combustion, and the accompanying pressure peaks in the cylinder, takes place shortly after TDC. Along with increasing engine speed, therefore, the ignition angle must be shifted in the advance direction.

The cylinder charge (or fill) also has an effect upon the combustion curve. The lower the cylinder charge the slower is the flame front's propagation. For this reason, with a low cylinder charge, the ignition angle must also be advanced.

1 Ignition map based on engine rpm *n* and relative air charge *rl*



Influence of the ignition angle

The ignition angle has a decisive influence on engine operation. It determines

- The delivered torque,
- The exhaust-gas emissions, and
- The fuel consumption.

The ignition angle is chosen so that all requirements are complied with as well as possible, whereby care must be taken that continued engine knock is avoided.

Ignition angle: Basic adaptation

On electronically controlled ignition systems, the ignition map (Fig. 1) takes into account the influence of engine speed and cylinder charge on the ignition angle. This map is stored in the engine-management data storage, and represents the basic adaptation of the ignition angle.

The x and y axes represent the engine speed and the relative air charge. The map's data points are formed by a given number of values, typically 16. A certain ignition angle is allocated to each pair of variates so that the map has 256 (16x16) adjustable ignition-angle values. By applying linear interpolation between two data points, the number of ignition-angle values is increased to 4096.

Using the ignition-map principle for the electronic control of the ignition angle means that for every engine operating point it is possible to select the best-possible ignition angle.

These ignition maps are generated by running the engine on the engine dynamometer.

Additive ignition-angle adjustments

A lean A/F mixture is more difficult to ignite. This means that more time is needed before the main combustion point is reached. A lean A/F mixture must therefore be ignited sooner. The A/F ratio λ thus has an influence on the ignition angle.

The coolant temperature is a further variable which affects the choice of the ignition angle. Temperature-dependent ignition-angle corrections are therefore also necessary. Such corrections are stored in the data storage in the form of fixed values or characteristic curves (e.g. temperature-dependent correction). They shift the basic ignition angle by the stipulated amount in either the advance or retard direction.

Special ignition angle

There are certain operating modes, such as idle and overrun, which demand an ignition angle which deviates from those defined by the ignition map. In such cases, access is made to special ignition-angle curves stored in the data storage.

Knock control

Knock is a phenomenon which occurs when ignition takes place too early. Here, once regular combustion has started, the rapid pressure increase in the combustion chamber leads to the auto-ignition of the unburnt residual mixture which has not been reached by the flame front. The resulting abrupt combustion of the residual mixture leads to a considerable local pressure increase. This generates a pressure wave which propagates through the combustion chamber until it hits the cylinder wall. At low engine speeds and when the engine is not making too much noise, it is then audible as combustion knock. At high speeds, the engine noises blanket the combustion knock.

If knock continues over a longer period of time, the engine can be damaged by the pressure waves and the excessive thermal loading. To prevent knock on today's high-compression engines, no matter whether of the manifold-injection or direct-injection type, knock control belongs to the standard scope of the engine-management system. With this system, knock sensors detect the start of knock and the ignition angle is retarded at the cylinder concerned. To obtain the best-possible engine efficiency, therefore, the basic adaptation of the ignition angle (ignition map) can be located directly at the knock limit.

On direct-injection gasoline engines, combustion knock only takes place in homogeneous operation. There is no tendency for the engine to knock in the stratified-charge mode since there is no combustible mixture in the stratified charge at the combustion chamber's peripheral zones.

Dwell angle

The energy stored in the ignition coil is a function of the length of time current flows through the coil (energisation time). In order not to thermally overload the coil, the time required to generate the required ignition energy in the coil must be rigidly adhered to. The dwell angle refers to the crankshaft and is therefore speed-dependent.

The ignition-coil current is a function of the battery voltage, and for this reason the battery voltage must be taken into account when calculating the dwell angle.

The dwell-angle values are stored in a map, the x and y axes of which represent rpm and battery voltage.

Systems for cylinder-charge control

On a gasoline engine running with a homogeneous A/F mixture, the intake air is the decisive quantity for the output torque and therefore for engine power. This means that not only is the fuel-metering system of special importance but also the systems which influence the cylinder charge. Some of these systems are able to influence the percentage of inert gas in the cylinder charge and thus also the exhaust emissions.

Air-charge control

For it to burn, fuel needs oxygen which the engine takes from the intake air. On engines with external A/F-mixture formation (manifold injection), as well as on direct-injection engines operating on a homogeneous A/F mixture with $\lambda = 1$, the output torque is directly dependent upon the intake-air mass. The throttle valve located in the induction tract controls the air flow drawn in by the engine and thus also the cylinder charge.

Conventional systems

Conventional systems (Fig. 1) feature a mechanically operated throttle valve (3). The accelerator-pedal (1) movement is transferred to the throttle valve by a linkage (2) or by a Bowden cable. The throttle valve's variable opening angle alters the opening cross-section of the intake passage (4) and in doing so regulates the air flow (5) drawn in by the engine, and with it the torque output.

To compensate for the higher levels of friction, the cold engine requires a larger air mass and extra fuel. And when, for instance, the A/C compressor is switched on more air is needed to compensate for the torque loss. This information is inputted to the ECU (8) in the form of an electrical signal (9), and the extra air is supplied by the air bypass actuator (7) directing the required extra air (6) around the throttle valve. Another method uses a throttle-valve actuator to adjust the throttle valve's minimum stop. In both cases though, it is only possible to electronically influence the air flow needed by the engine to a limited extent, for instance for idle-speed control.

1 Principle of the air control in a conventional system using a mechanically adjustable throttle valve and an air bypass actuator

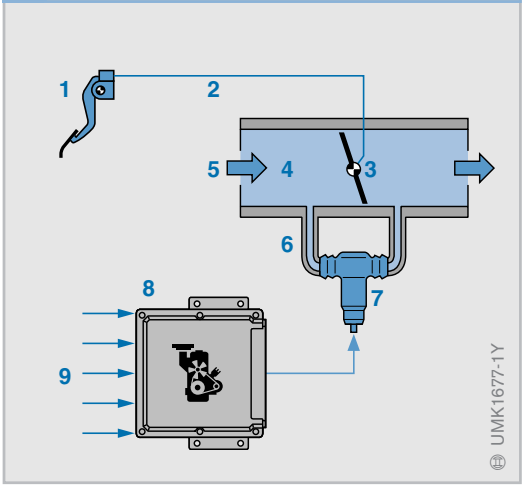


Figure 1
1 Accelerator pedal
2 Bowden cable or linkage
3 Throttle valve
4 Induction passage
5 Intake air flow
6 Bypass air flow
7 Idle-speed actuator (air bypass actuator)
8 ECU
9 Input variables (electrical signals)

Robert Bosch GmbH

2 The ETC system (Electronic Throttle Control or EGAS)

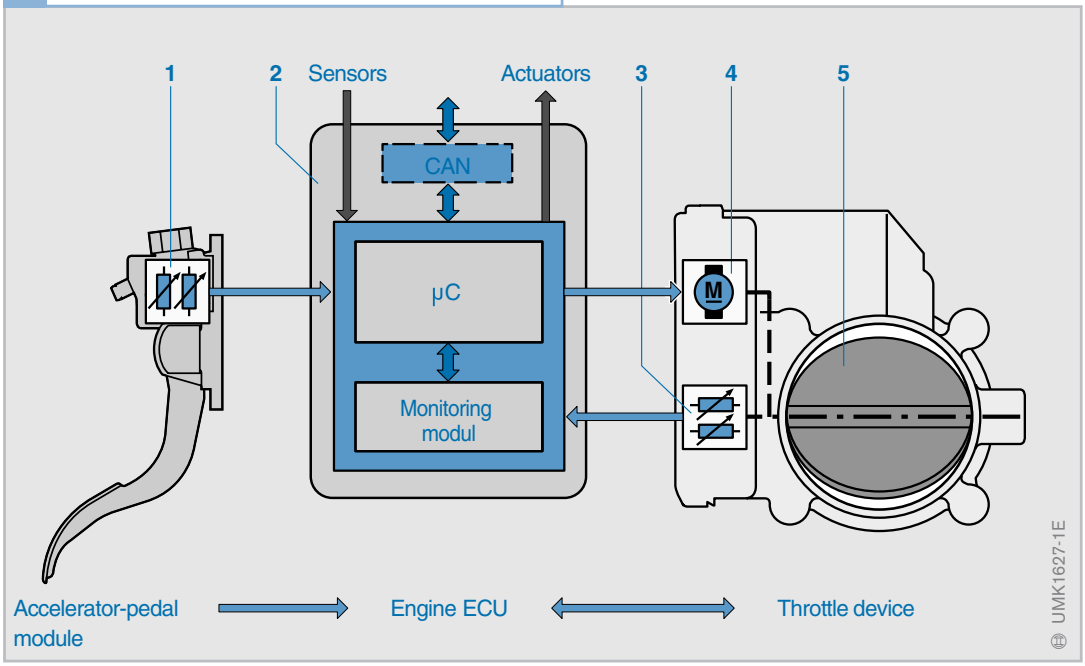


Figure 2
1 Accelerator-pedal sensor
2 Engine ECU
3 Throttle-valve-angle sensor
4 Throttle-valve drive (DC motor)
5 Throttle valve

ETC systems

With ETC (Electronic Throttle Control, also known as EGAS), an ECU (Fig. 2, Pos. 2) is responsible for controlling the throttle valve (5). The DC-motor throttle-valve drive (4) and the throttle-valve-angle sensor (3) are combined with the throttle valve to form a unit, the so-called throttle device. To trigger the throttle device, the accelerator-pedal position, in other words the driver input, is registered by two potentiometers (accelerator-pedal sensor, 1). Taking into account the engine's actual operating status (engine speed, engine temperature, etc.) the engine ECU then calculates the throttle-valve opening which corresponds to the driver input and converts it into a triggering signal for the throttle-valve drive.

Using the feedback information from the throttle-valve-angle sensor regarding the current position of the throttle valve, it then becomes possible to precisely adjust the throttle valve to the required setting.
Two potentiometers on the accelerator-pedal and two on the throttle unit are a component part of the ETC monitoring concept. The potentiometers are duplicated for redundancy reasons. In case malfunctions are detected in that part of the system which is decisive for the engine's power output, the throttle valve is immediately shifted to a predetermined position (emergency or limp-home operation).

In the latest engine-management systems, the ETC control is integrated in the engine ECU which is also responsible for controlling ignition, fuel injection, and the auxiliary functions. There is no longer a separate ETC control unit.

The demands of emissions-control legislation are getting sharper from year to year. They can be complied with though thanks to ETC with its possibilities of further improving the A/F-mixture composition.

ETC is indispensable when complying with the demands made by gasoline direct injection on the overall vehicle system.

Variable valve timing

Apart from using the throttle-valve to throttle the flow of incoming fresh gas drawn in by the engine, there are several other possibilities for influencing the cylinder charge. The proportion of fresh gas and of residual gas can also be influenced by applying variable valve timing.

Of great importance for valve timing is the fact that the behaviour of the gas columns flowing into and out of the cylinders varies considerably as a function of engine speed or throttle-valve opening. With invariable valve timing, therefore, this means that the exhaust and refill cycle can only be ideal for one single engine operating range. Variable valve timing, on the other hand, permits adaptation to a variety of different engine speeds and cylinder charges. This has the following advantages:

- Higher engine outputs,
- Favorable torque curve throughout a wide engine-speed range,
- Reduction of toxic emissions,
- Reduced fuel consumption,
- Reduction of engine noise.

1 Camshaft adjustment

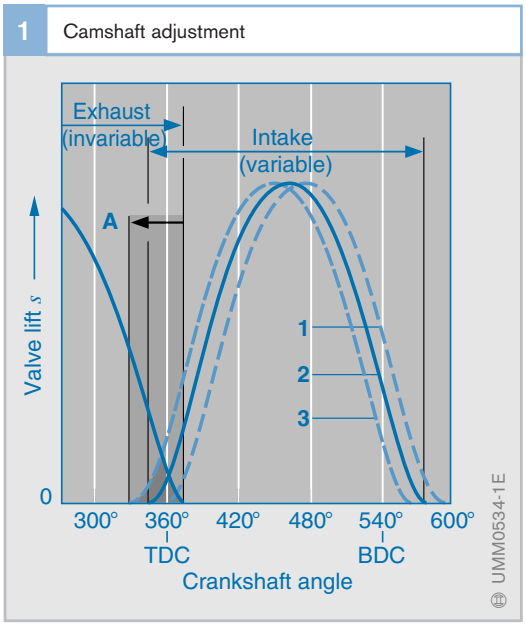


Figure 1
1 Camshaft retarded
2 Camshaft normal
3 Camshaft advanced
A Valve overlap

Camshaft phase adjustment

In conventional IC engines, camshaft and crankshaft are mechanically coupled to each other through toothed belt or chain. This coupling is invariable.

On engines with camshaft adjustment, at least the intake camshaft, but to an increasing degree the exhaust camshaft as well, can be rotated referred to the crankshaft so that valve overlap changes. The valve opening period and lift are not affected by camshaft phase adjustment, which means that "intake opens" and "intake closes" remain invariably coupled with each other.

The camshaft is adjusted by means of electrical or electro-hydraulic actuators. On less sophisticated systems provision is only made for two camshaft settings. Variable camshaft adjustment on the other hand permits, within a given range, infinitely variable adjustment of the camshaft referred to the crankshaft.

Fig. 1 shows how the "position", or lift, of the open intake-valve changes (referred to TDC) when the intake camshaft is adjusted.

Retard adjustment of the intake camshaft

Retarding the intake camshaft leads to the intake valve opening later so that valve overlap is reduced, or there is no valve overlap at all. At low engine speeds (<2000 min^{-1}), this results in only very little burnt exhaust gas flowing past the intake valve and into the intake manifold. At low engine speeds, the low residual exhaust-gas content in the intake of A/F mixture which then follows leads to a more efficient combustion process and a smoother idle. This means that the idle speed can be reduced, a step which is particularly favorable with respect to fuel consumption.

The camshaft is also retarded at higher engine speeds (>5,000 min^{-1}). Late closing of the intake valve, long after BDC, leads to a higher cylinder charge. This boost effect results from the high flow speed of the fresh gas through the intake valve which continues even after the piston has reversed its direction of travel and is moving upwards to compress the mixture. For this reason, the intake valve closes long after BDC.

Advance adjustment of the intake camshaft

In the medium speed range, the flow of fresh gas through the intake passage is much slower, and of course there is no high-speed boost effect.
At medium engine speeds, closing the intake valve earlier, only shortly after BDC, prevents the ascending piston forcing the freshly drawn-in gas out past the intake valve again and back into the manifold. At such speeds, advancing the intake camshaft results in better cylinder charge and therefore a good torque curve.

At medium speeds, advanced opening of the intake camshaft leads to increased valve overlap. Opening the intake valve early means that shortly before TDC, the residual exhaust gas which has not already left the cylinder is forced out past the open intake valve and into the intake manifold by the ascending piston. These exhaust gases are then drawn into the cylinder again and serve to increase the residual-gas content of the cylinder charge. The increased residual gas content in the freshly drawn in A/F mixture caused by advancing the intake camshaft, affects the combustion process. The resulting lower peak temperatures lead to a reduction in NO_x.

The higher inert-gas content in the cylinder charge makes it necessary to open the throttle valve further, which in turn leads to a reduction of the throttling losses. This means that valve overlap can be applied to reduce fuel consumption.

Adjusting the exhaust camshaft

On systems which can also adjust the exhaust camshaft, not only the intake camshaft is used to vary the residual-gas content, but also the exhaust camshaft. Here, the total cylinder charge (defined by "intake closes") and the residual-gas content (influenced by "intake opens" and "exhaust closes") can be controlled independently of each other.

Camshaft changeover

Camshaft changeover (Fig. 2) involves switching the camshaft between two different cam contours. This changes both the valve lift and the valve timing (cam-contour changeover). The first cam defines the optimum timing and the valve lift for the intake and exhaust valves in the lower and medium speed ranges. The second cam controls the increased valve lift and longer valve-open times needed at higher speeds.
At low and medium engine speeds, minimum valve lifts together with the associated

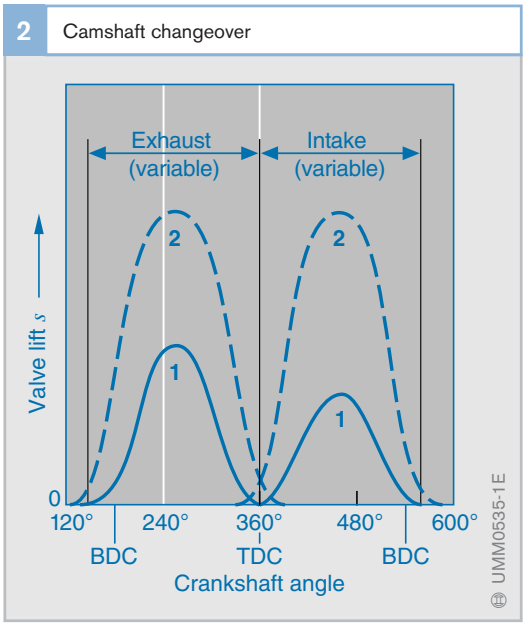


Figure 2
1 Standard cam
2 Supplementary cam

3 Example of a system with fully variable adjustment of valve timing and of valve lift

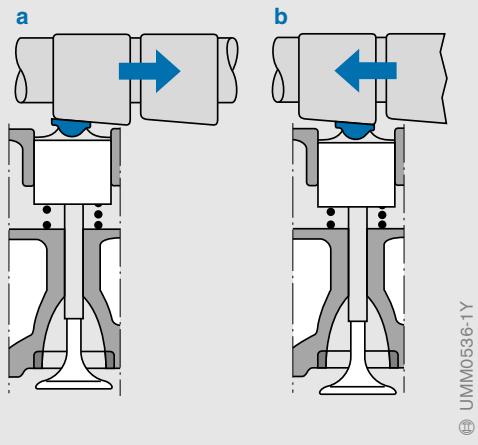


Figure 3
a Minimum lift
b Maximum lift

small valve-opening cross-sections lead to a high inflow velocity and therefore to high levels of turbulence in the cylinder for the fresh air (gasoline direct injection) or for the fresh A/F mixture (manifold injection). This ensures excellent A/F mixture formation at part load. The high engine outputs required at higher engine speeds and torque demand (WOT) necessitate maximum cylinder charge. Here, the maximum valve lift is selected.

There are a variety of methods in use for switching-over between the different cam contours. One method, for instance, relies on a free-moving drag lever which engages with the standard rocking lever as a function of rotational speed. Another method uses changeover cup tappets.

Fully variable valve timing and valve lift using the camshaft

Valve control which incorporates both variable valve timing and variable valve lift is referred to as being fully variable. Even more freedom in engine operation is permitted by 3D cam contours and longitudinal-shift camshafts (Fig. 3). With this form of camshaft control, not only the valve lift (only on the intake side) and thus the opening angle of the valves can be infinitely varied, but also the phase position between camshaft and crankshaft.

Since the intake valve can be closed early with this fully variable camshaft control, this permits so-called charge control in which the intake-manifold throttling is considerably reduced. This enables fuel consumption to be slightly lowered in comparison with the simple camshaft phase adjustment.

Fully variable valve timing and valve lift without using the camshaft

For valve timing, maximum design freedom and maximum development potential are afforded by systems featuring valve-timing control which is independent of the camshaft. With this form of timing, the valves are opened and closed, for instance, by electromagnetic actuators. A supplementary ECU is responsible for triggering. This form of fully variable valve timing without camshaft aims at extensive reduction of the intake-manifold throttling, coupled with very low pumping losses. Further fuel savings can be achieved by incorporating cylinder and valve shutoff.

These fully variable valve-timing concepts not only permit the best-possible cylinder charge and with it a maximum of torque, but they also ensure improved A/F-mixture formation which results in lower toxic emissions in the exhaust gas.

Exhaust-gas recirculation (EGR)

The mass of the residual gas remaining in the cylinder, and with it the inert-gas content of the cylinder charge, can be influenced by varying the valve timing. In this case, one refers to "internal" EGR. The inert-gas content can be influenced far more by applying "external" EGR with which part of the exhaust gas which has already left the cylinder is directed back into the intake manifold through a special line (Fig. 1, Pos. 3). EGR leads to a reduction of the NO_x emissions and to a slightly lower fuel-consumption figure.

Limiting the NO_x emissions

Since they are highly dependent upon temperature, EGR is highly effective in reducing NO_x emissions. When peak combustion temperature is lowered by introducing burnt exhaust gas to the A/F mixture, NO_x emissions drop accordingly.

Lowering fuel consumption

When EGR is applied, the overall cylinder charge increases while the charge of fresh air remains constant. This means that the throttle valve (2) must reduce the engine throttling if a given torque is to be achieved. Fuel consumption drops as a result.

EGR: Operating concept

Depending upon the engine's operating point, the engine ECU (4) triggers the EGR valve (5) and defines its opened cross-section. Part of the exhaust-gas (6) is diverted via this opened cross-section (3) and mixed with the incoming fresh air. This defines the exhaust-gas content of the cylinder charge.

EGR with gasoline direct injection

EGR is also used on gasoline direct-injection engines to reduce NO_x emissions and fuel consumption. In fact, it is absolutely essential since with it NO_x emissions can be lowered to such an extent in lean-burn operations that other emissions-reduction measures can be reduced accordingly (for instance, rich homogeneous operation for NO_x "Removal" from the NO_x accumulator-type catalytic converter). EGR also has a favorable effect on fuel consumption.

There must be a pressure gradient between the intake manifold and the exhaust-gas tract in order that exhaust gas can be drawn in via the EGR valve. At part load though, direct-injection engines are operated practically unthrottled. Furthermore a considerable amount of oxygen is drawn into the intake manifold via EGR during lean-burn operation.

Non-throttled operation and the introduction of oxygen into the intake manifold via the EGR therefore necessitate a control strategy which coordinates throttle valve and EGR valve. This results in severe demands being made on the EGR system with regard to precision and reliability, and it must be robust enough to withstand the deposits which accumulate in the exhaust-gas components as a result of the low exhaust-gas temperatures.

1 Exhaust-gas recirculation (EGR)

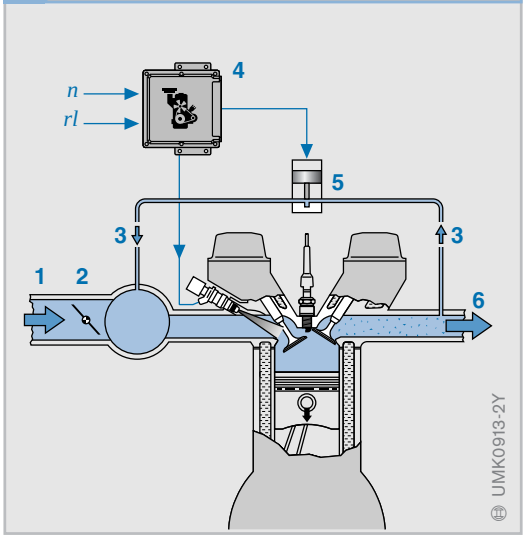


Figure 1
1 Fresh-air intake
2 Throttle valve
3 Recirculated exhaust gas
4 Engine ECU
5 EGR valve
6 Exhaust gas
n Engine rpm
rl Relative air charge

Dynamic supercharging

Approximately speaking, the achievable engine torque is proportional to the fresh-gas content in the cylinder charge. This means that the maximum torque can be increased to a certain extent by compressing the air before it enters the cylinder.

The exhaust-and-refill processes are not only influenced by the valve timing, but also by the intake and exhaust lines. The piston's induction work causes the open intake valve to trigger a return pressure wave. At the open end of the intake manifold, the pressure wave encounters the quiescent ambient air from which it is reflected back again so that it returns in the direction of the intake valve. The resulting pressure fluctuations at the intake valve can be utilized to increase the fresh-gas charge and thus achieve the highest-possible torque.

This supercharging effect thus depends on utilization of the incoming air's dynamic response. In the intake manifold, the dynamic effects depend upon the geometrical relationships in the intake manifold and on the engine speed.

For the even distribution of the A/F mixture, the intake manifolds for carburetor engines and single-point injection (TBI) must have short pipes which as far as possible must be of the same length for all cylinders. In the case of multipoint injection (MPI), the fuel is either injected into the intake manifold onto the intake valve (manifold injection), or it is injected directly into the combustion chamber (gasoline direct injection). With MPI, since the intake manifolds transport mainly air and practically no fuel can deposit on the manifold walls, this provides wide-ranging possibilities for intake-manifold design. This is the reason for there being no problems with multipoint injection systems regarding the even distribution of fuel.

Ram-tube supercharging

The intake manifolds for multipoint injection systems are composed of the individual tubes or runners and the manifold chamber.

In the case of ram-tube supercharging (Fig. 1), each cylinder is allocated its own tube (2) of specific length which is usually attached to the manifold chamber (3). The pressure waves are able to propagate in the individual tubes independently.

The supercharging effect depends upon the intake-manifold geometry and the engine speed. For this reason, the length and diameter of the individual tubes is matched to the valve timing so that in the required speed range a pressure wave reflected at the end of the tube is able to enter the cylinder through the open intake valve (1) and improve the cylinder charge. Long, narrow tubes result in a marked supercharging effect at low engine speeds. On the other hand, short, large-diameter tubes have a positive effect on the torque curve at higher engine speeds.

1 Principle of ram-tube supercharging

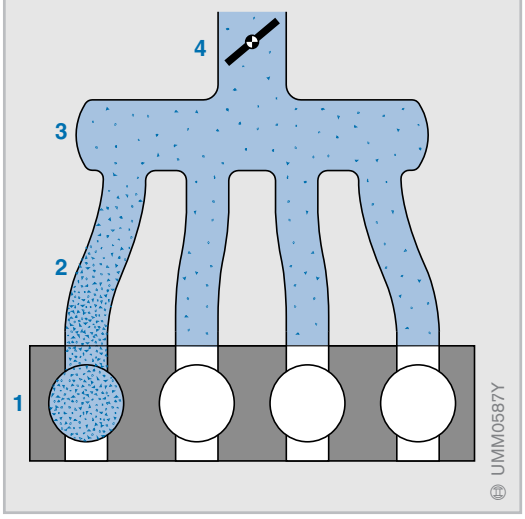


Figure 1
1 Cylinder
2 Individual tube
3 Manifold chamber
4 Throttle valve

Tuned-intake-tube charging

At a given engine speed, the periodic piston movement causes the intake-manifold gas-column to vibrate at resonant frequency. This results in a further increase of pressure and leads to an additional supercharging effect. On the tuned intake-tube system (Fig. 2), groups of cylinders (1) with identical angular ignition spacing are each connected to a resonance chamber (3) through short tubes (2). The chambers, in turn, are connected through tuned intake tubes (4) with either the atmosphere or with the manifold chamber (5) and function as Helmholtz resonators.

The subdivision into two groups of cylinders each with its own tuned intake tube prevents the overlapping of the flow processes of two neighboring cylinders which are adjacent to each other in the firing sequence.

The length of the tuned intake tubes and the size of the resonance chamber are a function of the speed range in which the supercharging effect due to resonance is required to be at maximum. Due to the accumulator effect of the considerable chamber volumes which are sometimes needed, dynamic-response errors can occur in some cases when the load is changed abruptly.

Variable-geometry intake manifold

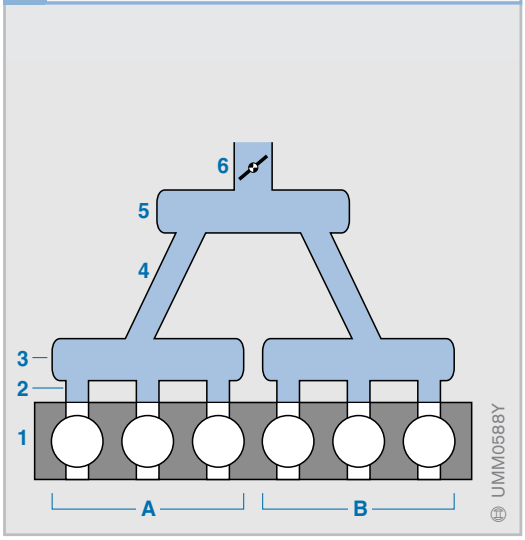
The supplementary cylinder charge resulting from dynamic supercharging depends upon the engine's working point. The two systems just dealt with increase the achievable maximum charge (volumetric efficiency), above all in the low engine-speed range (Fig. 3).

Practically ideal torque characteristics can be achieved with variable-geometry intake manifolds in which, as a function of the engine operating point, flaps are used to implement a variety of different adjustments such as:

- Adjustment of the intake-tube length,
- Switch over between different intake-tube lengths or different tube diameters,
- Selected switchoff of one of the cylinder's intake tubes on multiple-tube systems,
- Switchover to different chamber volumes.

Electrical or electropneumatically actuated flaps are used for change-over operations in these variable-geometry systems.

2 Principle of tuned-intake-tube charging



3 Increasing the maximum-possible cylinder air charge (volumetric efficiency) by means of dynamic supercharging

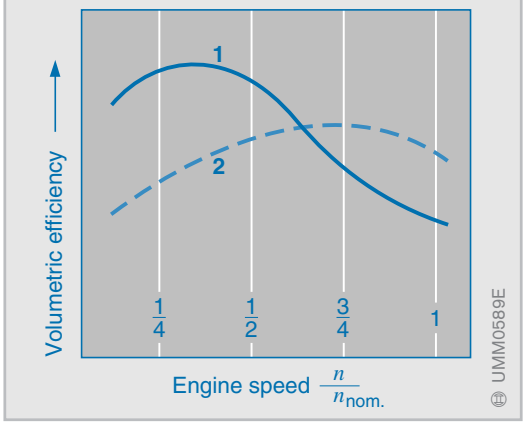


Figure 2
1 Cylinder
2 Short tube
3 Resonance chamber
4 Tuned intake tube
5 Manifold chamber
6 Throttle valve
A Cylinder group A
B Cylinder group B

Figure 3
1 System with tuned-intake-tube charging
2 System with conventional intake manifold

Robert Bosch GmbH

Ram-tube systems

The manifold system shown in Fig. 4 can switch between two different ram tubes. In the lower speed range, the changeover flap (1) is closed and the intake air flows to the cylinders through the long ram tube (3). At higher speeds and with the changeover flap open, the intake air flows through the short, wide diameter ram tube (4), and thus contibutes to improved cylinder charge at high engine revs.

Tuned-intake-tube system

Opening the resonance flap switches in a second tuned intake tube. The changed geometry of this configuration has an effect upon the resonant frequency of the intake system. Cylinder charge in the lower speed range is improved by the higher effective volume resulting from the second tuned intake pipe.

Combined tuned-intake-tube and ram-tube system

When design permits the open changeover flap (Fig. 5, Pos. 7) to combine both the resonance chambers (3) to form a single volume, one speaks of a combined tuned-intake-tube and ram-tube system. A single intake-air chamber with a high resonant frequency is then formed for the short ram tubes (2).

At low and medium engine revs, the changeover flap is closed and the system functions as a tuned-intake-tube system. The low resonant frequency is then defined by the long tuned intake tube (4).

4 Ram-tube system

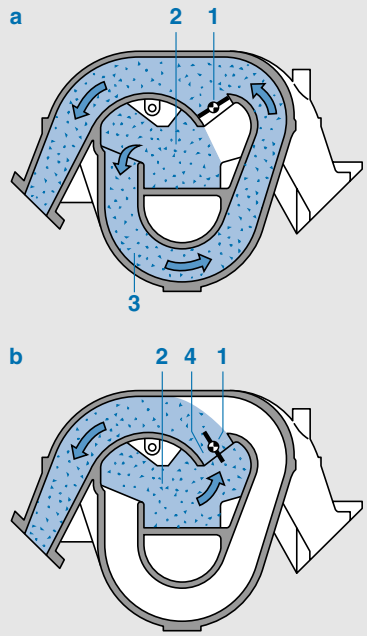


Figure 4
- a Manifold geometry with changeover flap closed
- b Manifold geometry with changeover flap open
- 1 Changeover flap
- 2 Manifold chamber
- 3 Changeover flap closed: Long, narrow-diameter ram tube
- 4 Changeover flap opened: Short, wide-diameter ram tube

5 Combined tuned-intake-tube and ram-tube system

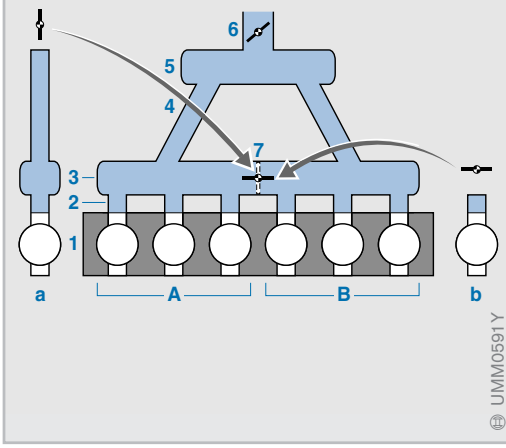


Figure 5
- 1 Cylinder
- 2 Ram tube (short intake tube)
- 3 Resonance chamber
- 4 Tuned intake tube
- 5 Manifold chamber
- 6 Throttle valve
- 7 Changeover flap
- A Cylinder group A
- B Cylinder group B

- a Intake-manifold conditions with changeover flap closed
- b Intake-manifold conditions with changeover flap open

Volkswagen Technical Site: http://vwts.ru http://vwts.info

Mechanical supercharging

Design and operating concept

The application of supercharging units leads to increased cylinder charge and therefore to increased torque. Mechanical supercharging uses a compressor which is driven directly by the IC engine. Mechanically driven compressors are either positive-displacement superchargers with different types of construction (e.g. Roots supercharger, sliding-vane supercharger, spiral-type supercharger, screw-type supercharger), or they are centrifugal turbo-compressors (e.g. radial-flow compressor). Fig. 1 shows the principle of functioning of the rotary-screw supercharger with the two counter-rotating screw elements. As a rule, engine and compressor speeds are directly coupled to one another through a belt drive.

Boost-pressure control

On the mechanical supercharger, a bypass can be applied to control the boost pressure. A portion of the compressd air is directed into the cylinder and the remainder is returned to the supercharger input via the bypass. The engine management is responsible for controlling the bypass valve.

Advantages and disadvantages

On the mechanical supercharger, the direct coupling between compressor and engine crankshft means that when engine speed increases there is no delay in supercharger acceleration. This means therefore, that compared to exhaust-gas turbocharging engine torque is higher and dynamic response is better.

Since the power required to drive the compressor is not available as effective engine power, the above advantage is counteracted by a slightly higher fuel-consumption figure compared to the exhaust-gas turbocharger. This disadvantage though is somewhat alleviated when the engine management is able to switch off the compressor via a clutch at low engine loading.

1 Rotary-screw supercharger: Principle of functioning

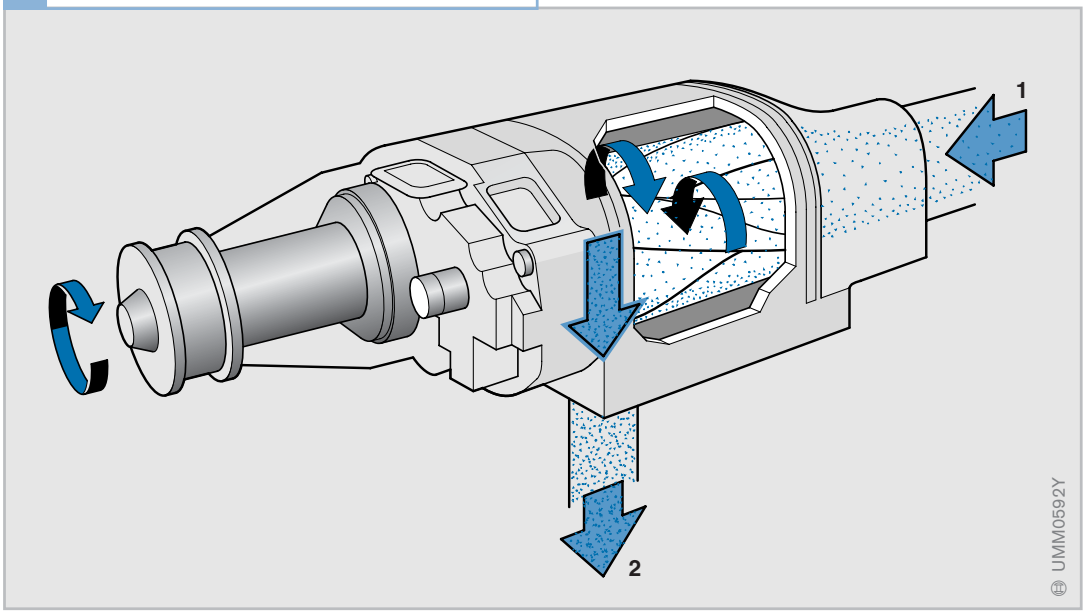


Figure 1
1 Intake air
2 Compressed air

Exhaust-gas turbocharging

Of all the possible methods for supercharging the IC engine, exhaust-gas turbocharging is the most widely used. Even on engines with low swept volumes, exhaust-gas supercharging leads to high torques and power outputs together with high levels of engine efficiency.

Whereas, in the past, exhaust-gas turbocharging was applied in the quest for increased power-weight ratio, it is today mostly used in order to increase the maximum torque at low and medium engine speeds. This holds true particularly in combination with electronic boost-pressure control.

Design and operating concept

The main components of the exhaust-gas turbocharger (Fig. 1) are the exhaust-gas turbine (3) and the compressor (1). The compressor impeller and the turbine rotor are mounted on a common shaft (2).

The energy needed to drive the exhaust-gas turbine is for the most part taken from the hot, pressurized exhaust gas. On the other hand, energy must be also used to "dam" the exhaust gas when it leaves the engine so as to generate the required compressor power.

The hot gases (Fig. 2, Pos. 7) are applied radially to the exhaust-gas turbine (4) and cause this to rotate at very high speed. The turbine-rotor blades are inclined towards the center and thus direct the gas to the inside from where it then exits axially.

1 Passenger-car exhaust-gas turbocharger (Shown: 3K-Warner, type K14)

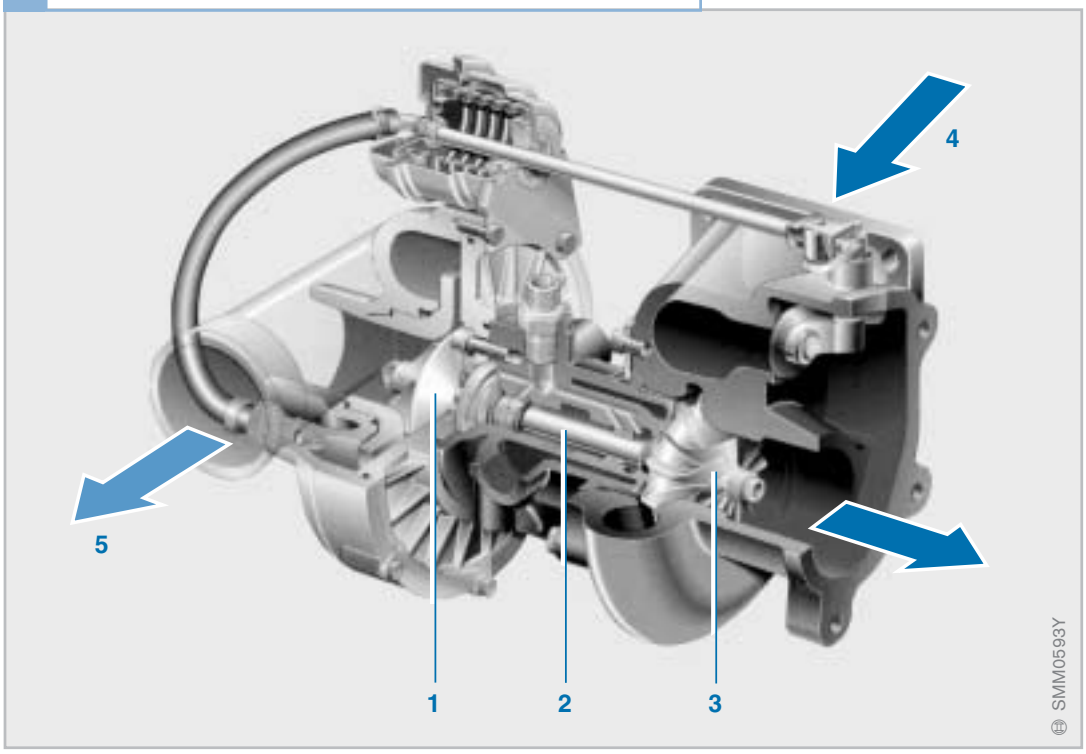


Figure 1
1 Compressor impeller
2 Shaft
3 Exhaust-gas turbine
4 Intake for exhaust-gas mass flow
5 Outlet for compressed air

The compressor (3) also turns along with the turbine, but here the flow conditions are reversed. The fresh incoming gas (5) enters axially at the center of the compressor and is forced radially to the outside by the blades and compressed in the process.

Since the exhaust-gas turbocharger is located directly in the flow of hot exhaust gas it must be built of highly temperature-resistant materials.

Exhaust-gas turbochargers: Designs

Wastegate supercharger

The objective is for IC engines to develop high torques at low engine speeds. The turbine casing has therefore been designed for a low level of exhaust-gas mass flow, for instance WOT at ≤ 2000 min^{-1}. With high exhaust-gas mass flows in this range, part of the flow must be diverted around the turbine and into the exhaust system in order that the turbocharger is prevented from overcharging the engine. Diversion is via a bypass valve, the so-called wastegate (Fig. 2, Pos. 8). This flap-type bypass valve is usually integrated into the turbine casing.

The wastegate is actuated by the boost-pressure control valve (6). This valve is connected pneumatically to the pulse valve (1) through a control line (2). The pulse valve changes the boost pressure upon being triggered by an electrical signal from the engine ECU. This electrical signal is a function of the current boost pressure, information on which is provided by the boost-pressure sensor (BPS).

If the boost pressure is too low, the pulse valve is triggered so that a somewhat lower pressure prevails in the control line. The boost-pressure control valve then closes the wastegate and the proportion of the exhaust-gas mass flow used to power the turbine is increased.

If, on the other hand, the boost pressure is excessive, the pulse valve is triggered so that a somewhat higher pressure is built up in the control line. The boost-pressure control valve then opens the wastegate and the proportion of the exhaust-gas mass flow used to power the turbine is reduced.

VTG turbocharger

The VTG (Variable Turbine Geometry) is another method which can be applied to limit the exhaust-gas mass flow at higher engine speeds (Fig. 3, next page). The VTG supercharger is state-of-the-art on diesel engines, but has not yet become successful on gasoline engines due to the high thermal stressing resulting from the far hotter exhaust gases.

By varying the geometry, the adjustable guide vanes (3) adapt the flow cross-section, and with it the gas pressure at the turbine, to the required boost pressure. At low speeds, they open up a small cross-section so that the exhaust-gas mass flow in the turbine reaches a high speed and in doing so also brings the exhaust-gas turbine up to high speed (Fig. 3a).

2 Design and construction of an exhaust-gas turbocharger using a wastegate turbocharger as an example

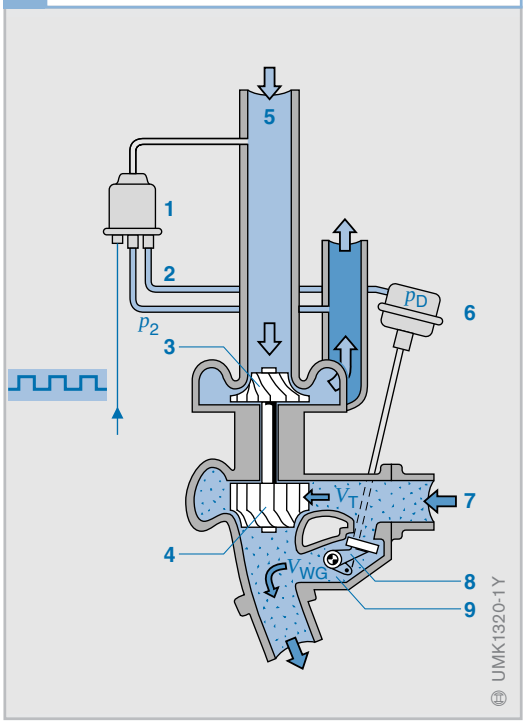


Figure 2

1 Pulse valve
2 Pneumatic control line
3 Compressor
4 Exhaust-gas turbine
5 Fresh incoming air
6 Boost-pressure control valve
7 Exhaust gas
8 Wastegate
9 Bypass duct
⎍⎍⎍ Triggering signal for pulse valve
V_T Volume flow through the turbine
V_{WG} Volume flow through the wastegate
p_2 Boost pressure
p_D Pressure on the valve diaphragm

Robert Bosch GmbH

At high engine speeds, the adjustable guide vanes (3) open up a larger cross-section so that more exhaust gas can enter without accelerating the exhaust-gas turbine to excessive speeds (Fig. 3b). This limits the boost pressure.

It is an easy matter to adjust the guide-vane angle by rotating the adjusting ring (2). Here, the guide vanes are adjusted to the desired angle either directly through individual adjusting levers (4) attached to the guide vanes, or by adjusting cam. The adjusting ring is rotated pneumatically via a barometric adjustment cell (5) using either vacuum or overpressure. This adjustment mechanism is triggered by the engine management so that the boost pressure can be set to the best-possible level in accordance with the engine's operating mode.

VST supercharger

On the VST (Variable Sleeve Turbine) supercharger, the "turbine size" is adapted by means of successively opening two flow passages (Fig. 4, Pos. 2 and 3) using a special control sleeve (4).

Initially, only one flow passage is opened, and the small opening cross-section results in high exhaust-gas flow speed and high turbine speeds (1). As soon as the permissible boost pressure is reached, the control sleeve successively opens the second flow passage, the exhaust-gas flow speed reduces accordingly, and with it the boost pressure.

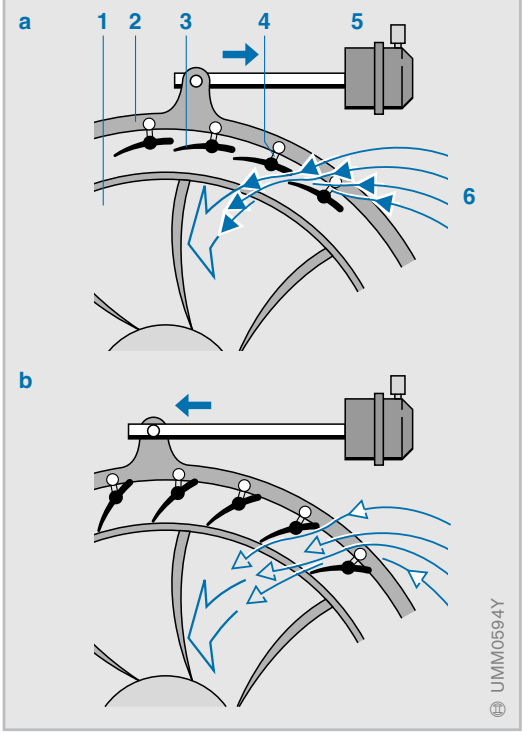
Using the bypass channel (5) incorporated in the turbine casing, it is also possible to divert part of the exhaust-gas mass flow past the exhaust-gas turbine.

The control sleeve is adjusted by the engine management via a barometric cell.

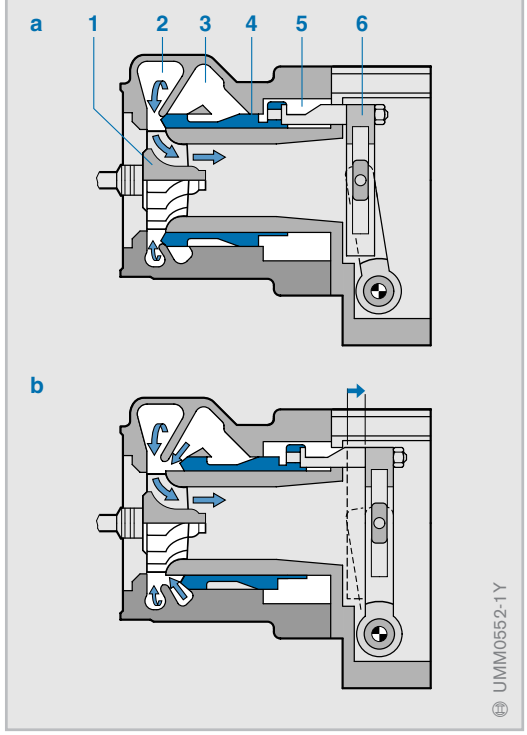
Figure 3
a Guide-vane setting for high boost pressure
b Guide-vane setting for low boost pressure
1 Exhaust-gas turbine
2 Adjusting ring
3 Guide vanes
4 Adjusting lever
5 Barometric cell
6 Exhaust-gas flow
◀ High flow speed
◁ Low flow speed

Figure 4
a Only 1 flow passage open
b Both flow passages open
1 Exhaust-gas turbine
2 1st flow passage
3 2nd flow passage
4 Special control sleeve
5 Bypass duct
6 Adjustment fork

3 Variable Turbine Geometry of the VTG supercharger



4 Turbine geometry of the VST supercharger



Robert Bosch GmbH

Exhaust-gas turbocharging: Advantages and disadvantages

Compared with a naturally-aspirated IC engine with the same output power, the major advantages are to be found in the turbocharged engine's lower weight and smaller size ("downsizing"). The turbocharged engine's torque characteristic is better throughout the usable speed range (Fig. 5, curve 4 compared to curve 3). All in all, at a given speed, this results in a higher output (A→B).

Due to its more favorable torque characteristic at WOT, the turbocharged engine generates the required power as shown in Fig. 5 (B or C) at lower engine speeds than the naturally aspirated engine. At part load, the throttle valve must be opened further, and the working point is shifted to an area with reduced frictional and throttling losses (C→B). This results in lower fuel-consumption figures even though turbocharged engines in fact feature less favorable efficiency figures due to their lower compression ratio.

The low torque that is available at very low engine speeds is a disadvantage of the turbocharger. In such speed ranges, there is not enough energy in the exhaust gas to drive the exhaust-gas turbine. In transient operation, even in the medium-speed range, the torque curve is less favorable than that of the natually aspirated engine (curve 5). This is due to the delay in building up the exhaust-gas mass flow. When accelerating from low engine speeds, this is evinced by the turbo flat spot.

The effects of this flat spot can be minimised by making full use of dynamic charge. This supports the supercharger's running-up characteristic. There are a number of other versions available, including a turbocharger with electric motor, or with an extra compressor driven by an electric motor. Independent of the exhaust-gas mass flow, these accelerate the compressor impeller and/or the air-mass flow, and in doing so avoid the turbo flat spot.

5 Power and torque characteristics of an exhaust-gas-turbocharged engine compared with those of a naturally aspirated engine

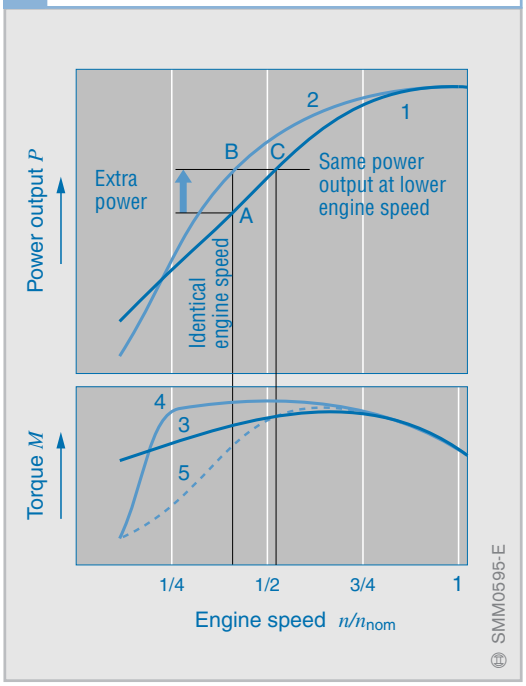


Figure 5
1, 3 Naturally aspirated engine in steady-state operation
2, 4 Supercharged engine in steady-state operation
5 Torque curve of the supercharged engine in transient (dynamic) operation

Intercooling

The air warms up in the compressor during the compression process, but since warm air has a lower density than cold air, this temperature rise has a negative effect upon cylinder charge. The compressed, warmed air must therefore be cooled off again by the intercooler. Compared to supercharged engines with this facility, intercooling results in an increase in the cylinder charge so that it is possible to further increase torque and output power.

The drop in the combustion-air temperature also leads to a reduction in the temperature of the cylinder charge compressed during the compression cycle. This has the following advantages:

- Reduced tendency to knock,
- Improved thermal efficiency resulting in lower fuel-consumption figures,
- Reduced thermal loading of the pistons,
- Lower NO_x emissions.

Gasoline fuel injection: An overview

It is the job of the fuel-injection system, or carburetor, to meter to the engine the best-possible air/fuel mixture for the actual operating conditions.

Fuel-injection systems, particularly when they are electronically controlled, are far superior to carburetors in complying with the tight limits imposed on A/F-mixture composition. In addition, they are better from the point of view of fuel consumption, driveability, and power output. In the automotive sector, the demands imposed by increasingly severe emission-control legislation have led to the carburetor being completely superseded by electronic fuel injection.

At present, on the majority of these injection systems the A/F mixture is formed externally outside the combustion chamber (manifold injection). Systems based on internal A/F-mixture formation, that is with the fuel injected directly into the cylinder (gasoline direct injection), are coming more and more to the forefront though, since they have proved to be particularly suitable in the never-ending endeavours to reduce fuel consumption.

1 Multipoint fuel-injection system

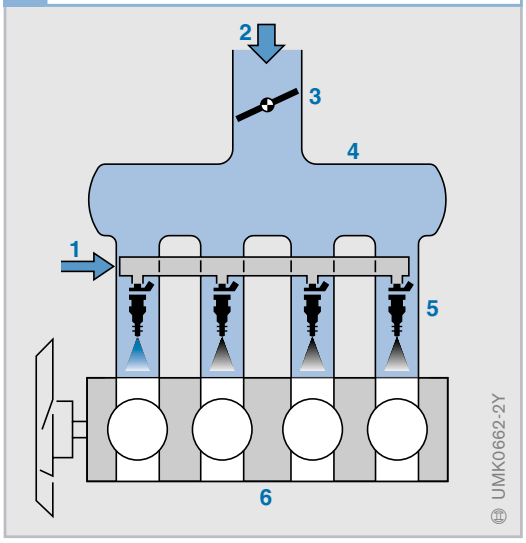


Figure 1
1 Fuel
2 Air
3 Throttle valve
4 Intake manifold
5 Injector
6 Engine

Overview

External A/F-mixture formation

On gasoline injection systems with external A/F-mixture formation, the mixture is formed outside the combustion chamber, that is, in the intake manifold. Development of such systems was forced ahead to enable them to comply with increasingly severe demands. Today, only the electronically controlled multipoint injection systems are of any importance in this sector.

Multipoint fuel-injection systems

On a multipoint injection system, every cylinder is allocated its own injector which sprays the fuel directly onto the cylinder's intake valve (Fig. 1). Such injection systems are ideal for complying with the demands made on the A/F-mixture formation system.

Mechanical fuel-injection system

The K-Jetronic injection system operates without any form of drive from the engine, and injects fuel continuously. The injected fuel mass is not defined by the injector but by the system's fuel distributor.

Combined mechanical-electronic fuel-injection system

The KE-Jetronic is based on the basic mechanical system used for the K-Jetronic. Thanks to additional operational-data acquisition, this system features electronically controlled supplementary functions which permit the injected fuel quantity to be even more accurately adapted to changing engine operating conditions.

Robert Bosch GmbH

Electronic fuel-injection systems
Electronically controlled fuel-injection systems inject the fuel intermittently through electromagnetically operated injectors. The injected fuel quantity is defined by the injector opening time (for a given pressure drop across the injector).

Examples: L-Jetronic, LH-Jetronic, and Motronic in the form of an integrated engine-management system (M and ME-Motronic).

Single-point injection

Single-point injection (also known as throttle-body injection or TBI) features an electromagnetically operated injector located at a central point directly above the throttle valve. This injection system intermittently injects fuel into the intake manifold (Fig. 2). The Bosch single-point injection systems are designated Mono-Jetronic and Mono-Motronic.

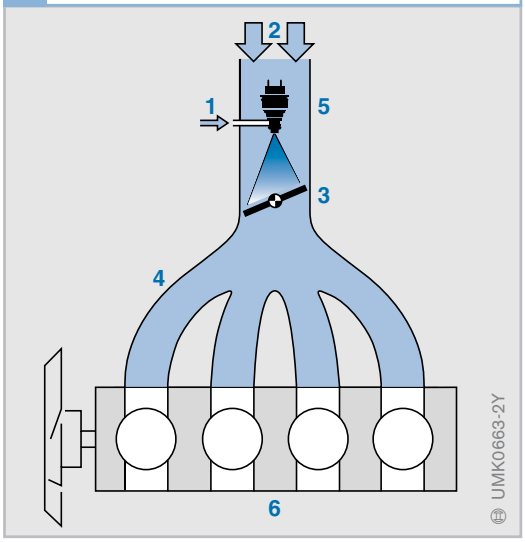
Internal A/F-mixture formation

On direct-injection (DI) systems, the fuel is injected directly into the combustion chamber through electromagnetic injectors, one of which has been allocated to each cylinder (Fig. 3).
A/F-mixture formation takes place inside the combustion chamber.

A/F-mixture formation inside the combustion chamber permits two completely different operating modes: In homogeneous operation, similar to external A/F-mixture formation, a homogeneous A/F mixture is present throughout the combustion chamber, and all the fresh air in the combustion chamber participates in the combustion process. This operating mode is therefore applied when high levels of torque are called for. In stratified-charge operation on the other hand, it is only necessary to have an ignitable A/F mixture around the spark plug. The remainder of the combustion chamber only contains fresh gas and residual gas without any unburnt-fuel content. This results in an extremely lean mixture at idle and part-load, with a corresponding drop in fuel consumption.

The MED-Motronic is used for the management of gasoline direct-injection engines.

2 Single-point injection (TBI) system



3 Direct-injection (DI) system

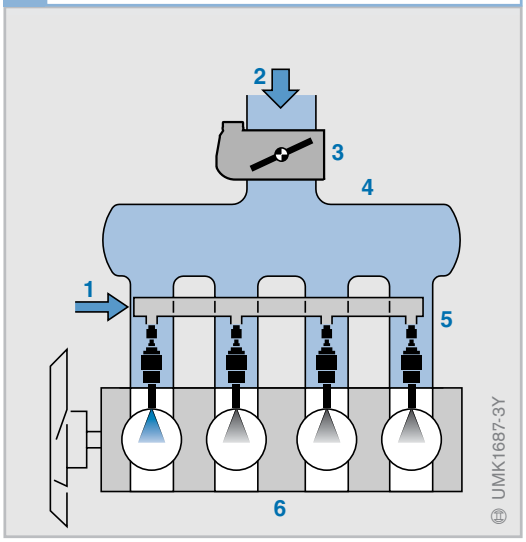


Figure 2
1 Fuel
2 Air
3 Throttle valve
4 Intake manifold
5 Injector
6 Engine

Figure 3
1 Fuel
2 Air
3 Throttle valve (ETC)
4 Intake manifold
5 Injectors
6 Engine

Fuel supply

The injectors (injection valves) of a gasoline injection system inject the fuel into the intake manifold (manifold injection), or directly into the combustion chamber (direct injection). In both cases, the fuel must be supplied to the injectors at a defined pressure.

This chapter describes the components which are involved in the supply of fuel from the fuel tank to the injectors or, in the case of gasoline direct injection, from the fuel tank to the high-pressure pump.

Overview

Basically speaking, the following components are mainly concerned with the supply of fuel as defined above (Fig. 1):

- Fuel tank (1),
- Electric fuel pump (2),
- Fuel filter (3),
- Fuel-pressure regulator (4), and
- Fuel lines (6 and 7).

With manifold injection, the fuel pump forces the fuel to the injector (8) via the fuel rail (5). On gasoline direct-injection engines, the fuel is forced into the high-pressure circuit by the high-pressure pump.

On older systems, the electric fuel pump is located outside the fuel tank in the fuel line itself (so-called "in-line" pump). On more recent systems, the fuel pump is inside the fuel tank ("in-tank" pump). It can also be combined with other components (e.g. preliminary filter, fuel-level sensor) in the tank in an in-tank unit.

The electric fuel pump delivers fuel continuously from the fuel tank and through the filter to the engine. The fuel-pressure regulator maintains a defined pressure in the fuel circuit, depending on the type of fuel-injection system.

In order that the required fuel pressure can be maintained under all operating conditions, the fuel pump delivers more fuel than is actually required by the engine. Excess fuel is returned to the tank.

So that the required fuel pressure is available for starting the engine, the electric fuel pump comes into operation immediately the ignition/starting switch is turned. If the engine is not started, it stops again after about 1 second.

To a great extent, the pressure generated by the fuel pump serves to prevent the formation of vapor bubbles in the fuel. The fuel system is provided with an integral non-return valve which decouples it from the fuel tank by preventing fuel returning to the tank. After the fuel pump has been switched off, the non-return valve maintains the system pressure for a certain period. This prevents the formation of vapor bubbles in the fuel system when the fuel heats up after the engine has been switched off.

Fuel supply for manifold injection

Here, there are two systems for fuel supply which differ according to the type of fuel return.

Fuel-supply system with fuel return

Excess fuel is that fuel which the injector does not inject (Fig 1, Pos. 8 and Fig. 2, next page, Pos. 8). It is returned to the fuel tank (1) via the fuel-pressure regulator (4) which is usually located on the fuel rail (5).

The intake-manifold pressure is applied as the reference for system-pressure control. Since the fuel-pressure regulator is situated very close to the manifold, it is possible here to locate the reference connection directly on the manifold. Using this reference pressure results in a constant difference between the fuel-system pressure and the intake-manifold pressure.

This has the advantage that the injected fuel quantity is a function of the injection time. It is independent of intake-manifold pressure and therefore also of cylinder charge.

Versions

There are a variety of different versions of the fuel-supply system with return. The standard version with fuel flowing through the rail is shown in Fig. 2a. There are also versions on the market in which the fuel line (6) in connected to the same end of the rail as the fuel-pressure regulator, so that there is no direct flow through the rail.

System pressure

On present-day systems with fuel return, the system pressure is approx. 0.3 MPa (3 bar).

1 Fuel supply system for a manifold-injection engine (version with fuel return)

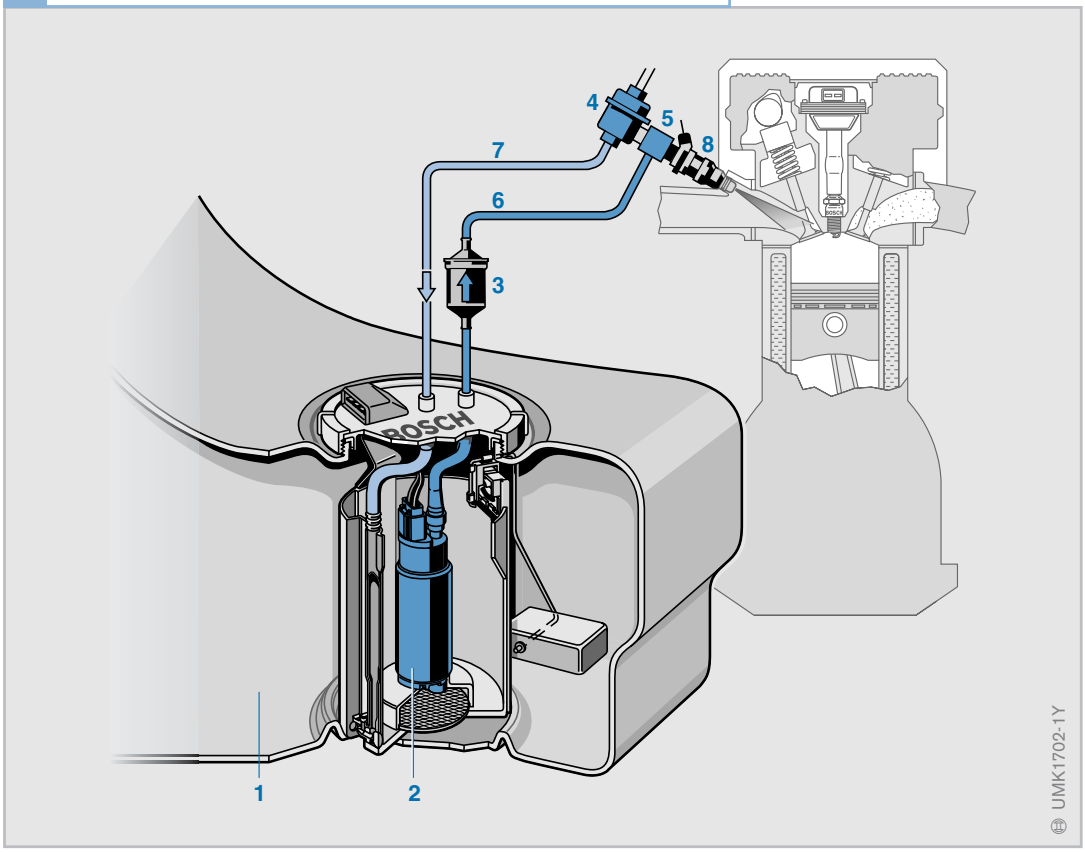


Figure 1

1 Fuel tank
2 Electric fuel pump (here integrated in the fuel tank),
3 Fuel filter,
4 Fuel-pressure regulator
5 Fuel rail
6 Fuel line
7 Fuel-return line
8 Injector

Robert Bosch GmbH

2 Fuel-supply system for a manifold-injection engine (examples)

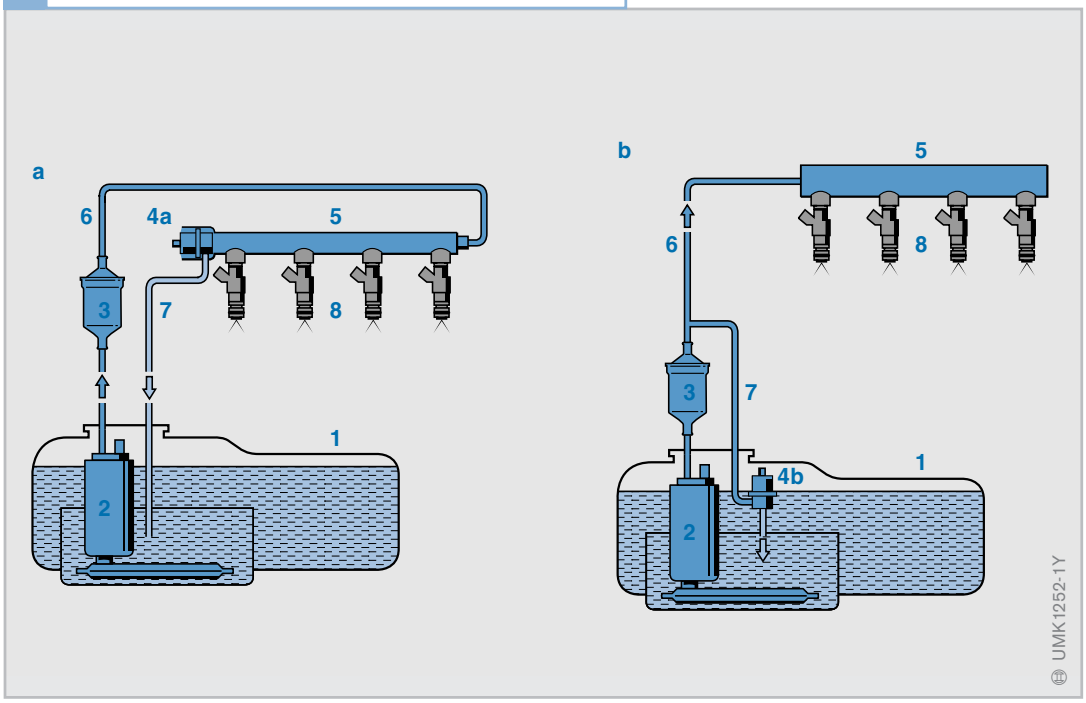


Figure 2
a With fuel return
b Without fuel return
1 Fuel tank
2 Electric fuel pump
3 Fuel filter
4a Fuel-pressure regulator (intake-manifold pressure used as reference)
4b Fuel-pressure regulator (surrounding pressure used as reference)
5 Fuel rail
6 Fuel line
7 Fuel-return line
8 Injectors

Returnless Fuel System

The fuel-pressure regulator (Fig. 2b, Pos. 4b) for the Returnless Fuel System (RLFS) is usually installed inside the fuel tank or in its vicinity. It can also be installed as a component part of the in-tank unit. On such systems, the fuel-return line from the fuel rail to the fuel tank can be dispensed with. The excess fuel delivered by the pump is returned directly to the tank via a short return line from the pressure regulator. Only the fuel actually injected by the injectors is delivered to the fuel rail.

This system has two advantages: Firstly lower costs, and secondly the fact that the fuel in the tank does not heat up since no hot fuel is returned from the engine compartment. This leads to a reduction in the HC emissions at the fuel tank, and therefore to reduced loading of the evaporative-emissions control system.

Versions

There are a number of different returnless fuel systems available:

- Fuel filter and pressure regulator outside the fuel tank,
- Fuel filter outside, pressure regulator inside the fuel tank,
- Fuel filter and pressure regulator both integrated in the in-tank unit (fuel-supply module).

System pressure

Since it would be too far away from the intake manifold it is practically impossible to provide an manifold reference connection at the fuel-pressure regulator. The fuel-pressure regulator therefore regulates the system pressure to a constant pressure differential referred to the surrounding/ambient pressure. This means that the injected fuel quantity is a function of the manifold pressure. This fact is taken into account when calculating the injection duration.

On returnless fuel systems the pressure is approx. 0.35...0.4 MPa (3.5...4 bar).

Low-pressure circuit for gasoline direct injection

On the gasoline direct-injection system, the fuel-supply system can be divided into the

- Low-pressure circuit, and
- High-pressure circuit.

The high-pressure circuit is described in the Chapter "Gasoline Direct Injection".

Depending upon the vehicle manufacturer's requirements, the low-pressure circuits for such injection systems can differ considerably in design. Similar to the manifold injection system, there are also variants here

- With fuel return, and
- Without fuel return (RLFS).

Example of an installation

Fig. 1 shows a fuel system featuring both fuel return and primary-pressure changeover. Here, the pressure in the low-pressure (primary pressure) circuit can be switched between two different levels.

Higher primary pressure

When the fuel is hot, measures must be taken to prevent the formation of vapor bubbles in the high-pressure pump (7) during the starting phase and the subsequent hot-idle phase. Increasing the primary pressure is a suitable step. Here, the shutoff valve (3) remains closed so that the pressure limiter integrated in the electric fuel pump (2) comes into operation and adjusts the primary pressure to 0.5 MPa (5 bar).

When located in the fuel tank, the pressure limiter not only protects the components against excess pressure, but also assumes responsibility for pressure-control functions.

Lower primary pressure

After 30...60 seconds, the high-pressure pump has been thoroughly flushed and cooled off far enough so that there is no longer any danger of vapor-bubble formation. The shutoff valve opens, and the pressure regulator (4) takes over the pressure-control function and adjusts the primary pressure to 0.3 MPa (3 bar).

In this case, the pressure regulator is located in the engine compartment. This is a fuel system with return.

1 Fuel supply for a gasoline direct-injection system (example with fuel return and primary-pressure changeover)

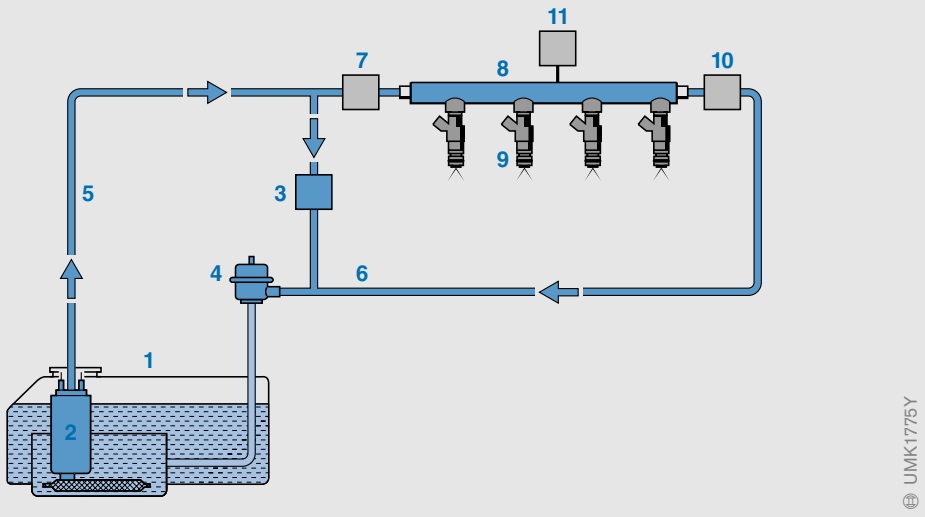


Figure 1
Low-pressure (primary) circuit with
1 Fuel tank
2 Electric fuel pump with integral pressure limiter and fuel filter
3 Shutoff valve
4 Pressure regulator
5 Fuel line
6 Fuel return line
High-pressure circuit with
7 High-pressure pump
8 Rail
9 High-pressure injectors
10 Pressure-control valve
11 Fuel-pressure sensor

Integration in the vehicle: In-tank unit

In the early years of electronically controlled gasoline injection, the electric fuel pump was always installed in the fuel line (“in-line”) outside the fuel tank. Today, on the other hand, the majority of electric fuel pumps are of the “in-tank” type and, as the name implies, are part of an “in-tank unit”, the so-called fuel-supply module. This contains an increasing number of other components, for instance:

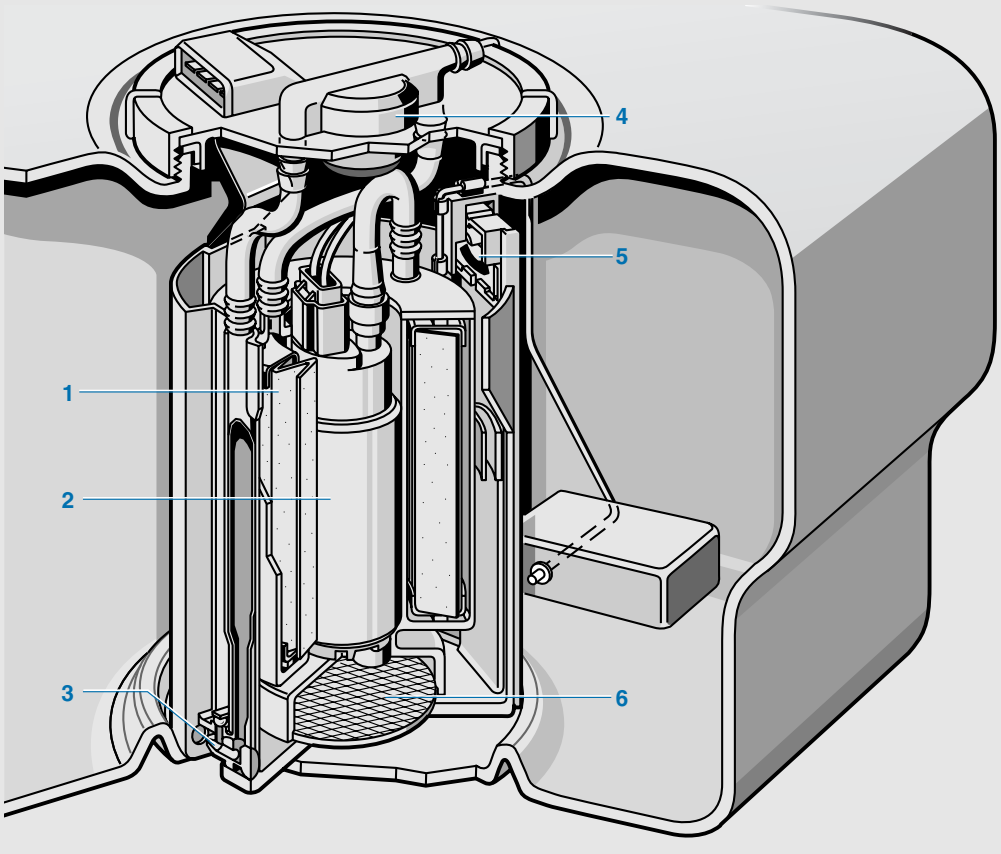
- A preliminary filter,
- A fuel-level sensor,
- Electric and hydraulic connections, and
- A special fuel reservoir for maintaining the fuel supply when cornering or in sharp bends.

Usually, a jet pump or a separate stage in the electric fuel pump keep this reservoir full.

On RLFS systems, the fuel-pressure regulator (4), is usually integrated in the in-tank unit where it is responsible for the fuel return. The pressure-side fine fuel filter can also be located in the in-tank unit.

In future, the fuel-supply module will incorporate further functions, for instance diagnosis devices for detecting tank leaks, or the timing module for triggering the electric fuel pump.

In-tank unit: The complete unit for a returnless fuel system (RLFS)



1 Fuel filter
2 Electric fuel pump
3 Jet pump (closed-loop controlled)
4 Fuel-pressure regulator
5 Fuel-level sensor
6 Preliminary filter

Evaporative-emissions control system

In order to comply with the legal limits for evaporative hydrocarbon emissions, vehicles are being equipped with evaporative-emissions control systems. This system prevents fuel vapor escaping to the atmosphere from the fuel tank.

Fuel-vapor generation

More fuel vapor escapes from the fuel tank under the following circumstances:

- When the fuel in the fuel tank warms up, due either to high surrrounding temperatures, or to the return to tank of excess fuel which has heated up in the engine compartment, and
- When the surrounding pressure drops, for instance when driving up a hill in the mountains.

Design and operating concept

The evaporative-emissions control system (Fig. 1) comprises the carbon canister (3), into which is led the venting line (2) from the fuel tank (1), together with the so-called canister-purge valve (5) which is connected to both the carbon canister and the intake manifold (8).

The activated carbon in the carbon canister absorbs the fuel contained in the fuel vapor and thus permits only air to escape into the atmosphere. As soon as the canister-purge valve opens the line (6) between the carbon canister and the intake manifold, the vacuum in the manifold causes fresh air to be drawn through the activated carbon. The absorbed fuel is then entrained with the fresh air (purging or regeneration of the activated carbon) and burnt in the normal combustion process. The system control reduces the injected fuel quantity by the amount returned through canister-purge valve. Regeneration is a closed-loop control process, whereby the fuel concentration in the canister-purge gas flow is continuously calculated based on the changes it causes in the excess-air factor λ.

The canister-purge gas quantity is controlled as a function of the working point and can be very finely metered using the canister-purge valve. In order to ensure that the carbon canister is always able to absorb fuel vapor, the activated carbon must be regenerated at regular intervals.

Gasoline direction injection: Special features

During stratified-charge operation on gasoline direct-injection engines, the possibility of regenerating the carbon canister's contents is limited due to the low level of vacuum in the intake manifold (caused by practically 100 % "unthrottled" operation) and the incomplete combustion of the homogeneously distributed canister-purge gas. This results in reduced canister-purge gas flow compared to homogeneous operation.
For instance, if the canister-purge gas flow is inadequate for coping with high levels of gasoline evaporation, the engine must be operated in the homogeneous mode until the high concentrations of gasoline in the canister-purge gas flow have dropped far enough.

1 Evaporative-emissions control system

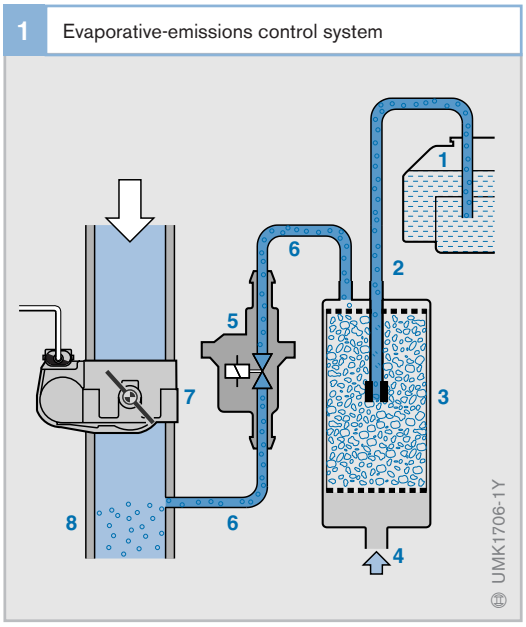


Figure 1
1 Fuel tank
2 Fuel-tank venting line
3 Carbon canister
4 Fresh air
5 Canister-purge valve
6 Line to the intake manifold
7 Throttle valve
8 Intake manifold

Robert Bosch GmbH

Electric fuel pump

Assignment

The electric fuel pump (EKP) must at all times deliver enough fuel to the engine at a high enough pressure to permit efficient fuel injection. The most important performance demands made on the pump are:

- Delivery quantity between 60 and 200 l/h at rated voltage,
- Pressure in the fuel system between 300 and 450 kPa (3...4.5 bar),
- System-pressure buildup even down to as low as between 50 and 60 % of rated voltage.

Apart from this, the EKP is increasingly being used as the pre-supply pump for the modern direct-injection systems used on diesel and gasoline engines.

On gasoline direct-injection systems for instance, pressures of up to 700 kPa are sometimes required during hot-delivery operations.

1 Electric fuel pump: Design and construction using a turbine pump as an example

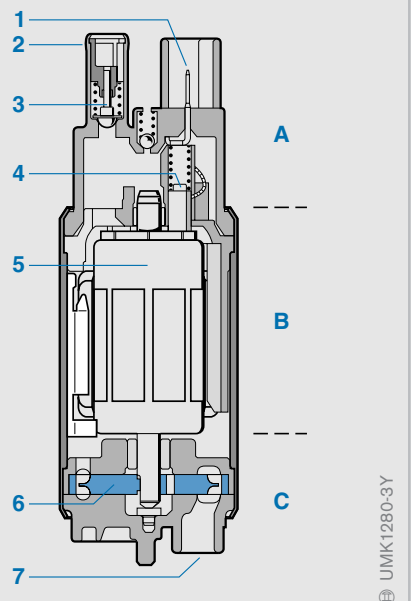


Figure 1
1 Electric connections
2 Hydraulic connections (fuel outlet)
3 Non-return valve
4 Carbon brushes
5 Permanent-magnet motor armature
6 Turbine-pump impeller ring
7 Hydraulic connection (fuel inlet)

Design and construction

The electric fuel pump is comprised of:

- End plate (Fig. 1, A), incorporating spark-suppression elements if required,
- Electric motor (B), and
- Pump element (C), designed as either positive-displacement or turbine pump (for description, see Section "Types" below).

Types

Positive-displacement pumps

In this type of pump, the fuel is drawn in, compressed in a closed chamber by rotation of the pump element, and transported to the high-pressure side. For the EKP, internal-gear pumps or roller-cell pumps (Figs. 2a, 2b) are used. When high system pressures are needed (400 kPa and above), positive-displacement pumps are particularly suitable. These feature a good low-voltage characteristic, that is, they have a relatively flat delivery-rate characteristic as a function of the operating voltage. Efficiency can be as high as 25 %.

Pressure pulsations, which are unavoidable, can cause audible noise depending upon the particular design details and installation conditions. The fact that the delivery rate can drop when the fuel is hot is another disadvantage which can occur in exceptional cases. This is due to vapor bubbles being pumped instead of fuel, and for this reason conventional positive-displacement pumps are equipped with peripheral preliminary stages for degassing purposes.

Whereas in electronic gasoline-injection systems the positive-displacement pump has to a great extent been superseded by the turbine pump for the classical fuel-pump requirements, it has captured a new field of application as the presupply pump on direct-injection systems wihich operate with far higher fuel-pressures.

Robert Bosch GmbH

Turbine pumps

This type of pump comprises an impeller ring with numerous blades inserted in slots around its periphery (Fig. 2c, Pos. 6). The impeller ring with blades rotates in a chamber formed from two fixed housing sections, each of which has a passage (7) adjacent to the blades which starts at the level of the intake port (A) and terminates where the fuel is forced out of the pump at system pressure through the fuel outlet (B). The "Stopper" between start and end of the passage prevents internal leakage.

At a given angle and distance from the intake opening a small degassing bore has been provided which provides for the exit of any gas bubbles which may be in the fuel. This, although improving the hot-delivery characteristics, is at the cost of very slight internal leakage. The degassing bore is not needed with diesel applications.

Pressure builds up along the passage (7) as a result of the exchange of pulses between the ring blades and the liquid particles. This leads to spiral-shaped rotation of the liquid volume trapped in the impeller ring and in the passages. In the case of the peripheral pump (Fig. 2c), the ring blades around the periphery of the ring are surrounded completely by the passage (hence the word "peripheral"). On the side-channel pump, the two channels are located on each side of the impeller ring adjacent to the blades.

Turbine pumps feature a low noise level since pressure buildup takes place continuously and is practically pulsation-free. Efficiency is between 10 % and about 20 %. Construction though is far simpler than that of the positive-displacement pumps.

Single-stage pumps can generate system pressures of up to 450 kPa. In future, turbine pumps will also be suitable for the higher system pressures that will be needed for brief periods on highly supercharged engines and gasoline direct-injection engines.

2 Principle of functioning of electric fuel pumps

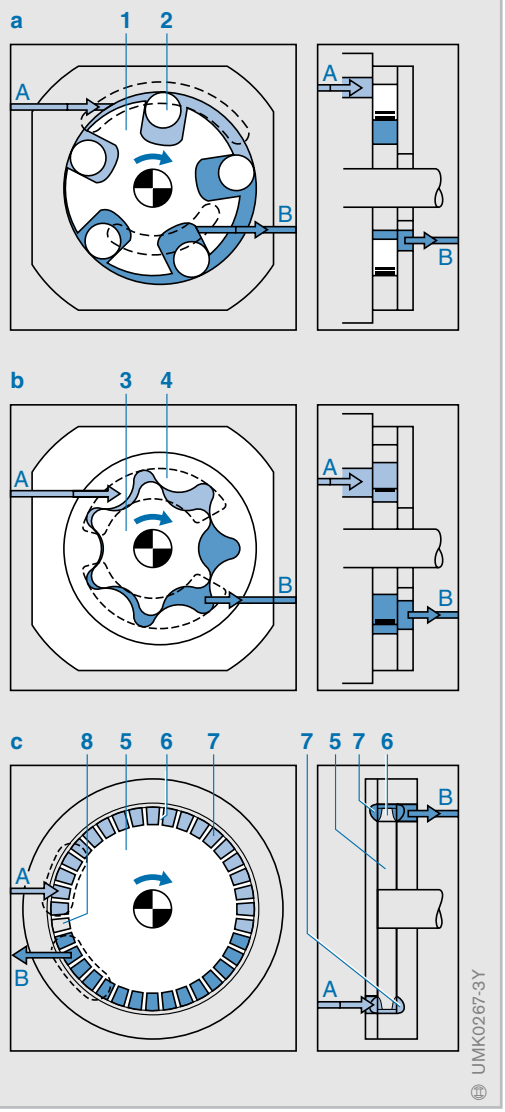


Figure 2
a Roller-cell pump (RZP)
b Inner-gear pump (IZP)
c Peripheral pump (PP)
A Intake port
B Outlet
1 Slotted rotor (eccentric)
2 Roller
3 Inner drive wheel
4 Rotor
5 Impeller ring
6 Impeller-ring blades
7 Passage (peripheral)
8 "Stopper"

For costs reasons, and due to their being quieter, turbine pumps are used almost exclusively on newly designed gasoline-engine automobiles.

Fuel filter

The injection systems for automobile spark-ignition (SI) engines operate with extreme precision. In order not to damage their precision parts, it is imperative that the fuel is efficiently cleaned. Filters in the fuel circuit remove the solid particles which could cause wear. Such filters are either replaceable in-line filters, or are integrated in the fuel tank as "lifetime" in-tank filters. Apart from the filter's purely straining or filtering effect, a number of different processes are applied in order to remove the contaminants from the fuel. These include impact, diffusion, and blocking effects.

1 Section through a fuel filter

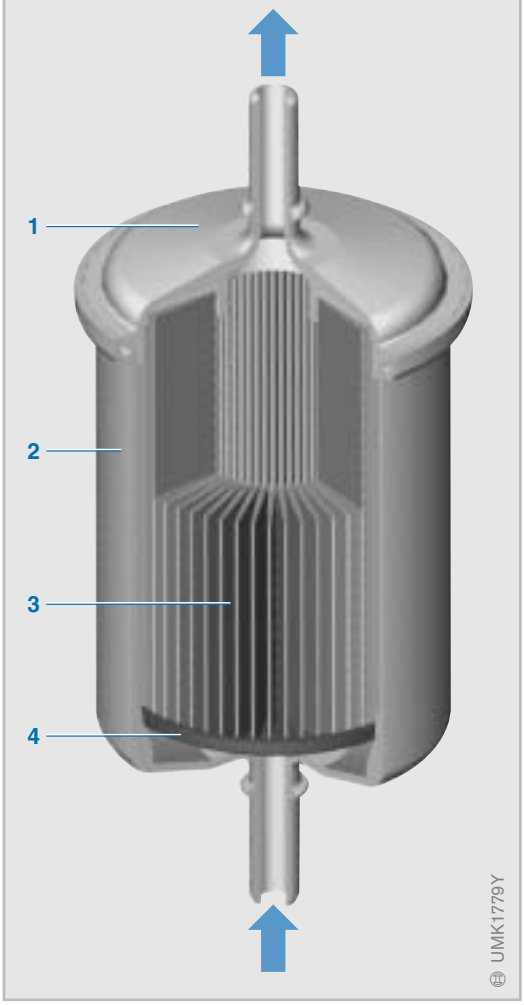


Figure 1
1 Filter cover
2 Filter housing
3 Filter element
4 Support plate

The filtration efficiency of the individual effects is a function of the size and the flow speed of the contaminant particles, and the filter medium is matched to these factors.

Pleated paper, which is sometimes specially impregnated, has come to the forefront as the filter medium (Fig. 1, Pos. 3). The filter medium is arranged in the fuel circuit so that the velocity of the fuel flow through all sections of its surface is as uniform as possible.

Whereas on manifold-injection systems the filter element has a mean pore size of 10 µm, far finer filtering is needed for gasoline direct-injection systems where up to 85 % of the particles larger than 5 µm must be reliably filtered out of the fuel.

In addition, for gasoline direct injection, when a new filter is fitted the traces of contaminant remaining in the filter after manufacture are an important factor: Metal, mineral, plastic, and glass-fiber particles must not exceed 200 µm.

Depending upon the filter volume, the useful life (guaranteed mileage) of the conventional in-line filter is somewhere between 37,500 and 55,000 miles (60,000...90,000 km). Guaranteed mileages of 100,000 miles (160,000 km) apply for in-tank filters. There are in-tank and in-line filters available for use with gasoline direct-injection systems which feature service lives in excess of 150,000 miles (250,000 km).

Filter housings (2) are either steel, aluminum, or plastic (100 % free from metal). Connections of the threaded, hose, or quick-connect type are used.

Filter efficiency depends on the throughflow direction. When replacing in-line filters, it is imperative that the flow direction given by the arrow is observed.

Fuel rail

Manifold injection

The fuel rail has the following assignments:

- Mounting and location of the injectors,
- Storage of the fuel volume,
- Ensuring that fuel is distributed evenly to all injectors.

In addition to the injectors, the fuel rail usually accomodates the fuel-pressure regulator and possibly even a pressure damper. Local fuel-pressure fluctuations caused by resonance when the injectors open and closed, is prevented by careful selection of the fuel-rail dimensions. As a result, irregularities in injected fuel quantity which can arise as a function of load and engine speed are avoided.

Depending upon the particular requirements of the vehicle in question, plastic or stainless-steel fuel rails are used. The fuel rail can incorporate a diagnosis valve for workshop testing purposes.

Gasoline direct injection

On gasoline DI systems, the rail is located downstream of the high-pressure pump, and is an integral part of the high-pressure stage.

Fuel-pressure regulator

Manifold injection

The amount of fuel injected by the injector (injected fuel quantity) depends upon the injection period and the difference between the fuel pressure in the fuel rail and the counterpressure in the manifold. On fuel systems with return, the influence of pressure is compensated for by a pressure regulator which maintains the difference between fuel pressure and manifold pressure at a constant level. This pressure regulator permits just enough fuel to return to the tank so that the pressure drop across the injectors remains constant. In order to ensure that the fuel rail is efficiently flushed, the fuel-pressure regulator is normally located at the end of the rail which leads the fuel tank.

1 Fuel-pressure regulator DR2

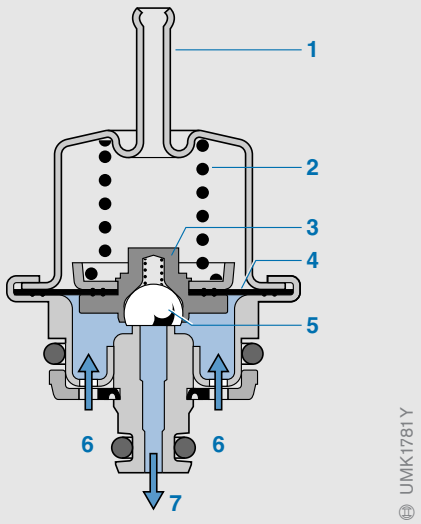


Figure 1
1 Intake-manifold connection
2 Spring
3 Valve holder
4 Diaphragm
5 Valve
6 Fuel inlet
7 Fuel return

On returnless fuel systems (RLFS), the pressure regulator is part of the in-tank unit installed in the fuel tank. The fuel-rail pressure is maintained at a constant level with reference to the surrounding pressure. This means that the difference between fuel-rail pressure and manifold pressure is not constant and must be taken into account when the injection duration is calculated.

The fuel-pressure regulator (Fig. 1) is of the diaphragm-controlled overflow type. A rubber-fabric diaphragm (4) divides the pressure regulator into a fuel chamber and a spring chamber. Through a valve holder (3) integrated in the diaphragm, the spring (2) forces a movable valve plate against the valve seat so that the valve closes. As soon as the pressure applied to the diaphragm by the fuel exceeds the spring force, the valve opens again and permits just enough fuel to flow back to the fuel tank that equilibrium of forces is achieved again at the diaphragm.

On multipoint fuel-injection systems, in order that the manifold vacuum can be applied to the spring chamber, this is connected pneumatically to the intake manifold at a point downstream of the throttle plate. There is therefore the same pressure ratio at the diaphragm as at the injectors. This means that the pressure drop across the injectors is solely a function of spring force and diaphragm surface area, and therefore remains constant.

Gasoline direct injection
On gasoline direct-injection systems, it is necessary to regulate the pressures in the high-pressure and the low-pressure stage, whereby the same fuel-pressure regulators are used for the low-pressure stage as for manifold injection.

Fuel-pressure damper

The repeated opening and closing of the injectors, together with the periodic supply of fuel when electric positive-displacement fuel pumps are used, leads to fuel-pressure oscillations. These can cause pressure resonances which adversely affect fuel-metering accuracy. It is even possible that under certain circumstances, noise can be caused by these oscillations being transferred to the fuel tank and the vehicle bodywork through the mounting elements of the fuel rail, fuel lines, and fuel pump.

These problems are alleviated by the use of special-design mounting elements and fuel-pressure dampers.The fuel-pressure damper is similar in design to the fuel-pressure regulator. Here too, a spring-loaded diaphragm separates the fuel chamber from the air chamber. The spring force is selected such that the diaphragm lifts from its seat as soon as the fuel pressure reaches its working range. This means that the fuel chamber is variable and not only absorbs fuel when pressure peaks occur, but also releases fuel when the pressure drops. In order to always operate in the most favorable range when the absolute fuel pressure fluctuates due to conditions at the manifold, the spring chamber can be provided with an intake-manifold connection.

Similar to the fuel-pressure regulator, the fuel-pressure damper can also be attached to the fuel rail or installed in the fuel line. In the case of gasoline direct injection, it can also be attached to the high-pressure pump.

Fuel tank

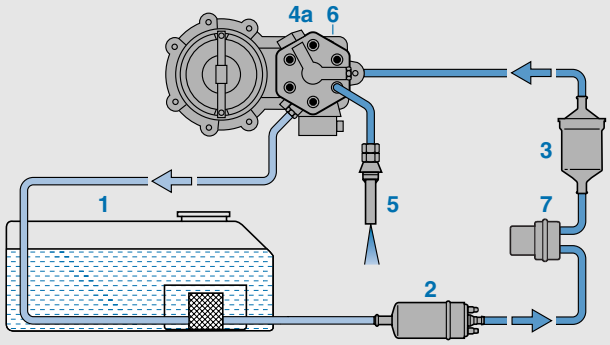
As its name implies, the fuel tank is used as the reservoir for the fuel. It must be non-corroding and must remain free of leaks at up to twice working pressure, or up to at least 0.03 MPa (0.3 bar) gauge pressure. Openings or safety valves must be provided for excess pressure to escape automatically. During cornering, on inclines, and in case of shock or impact, no fuel may leak out through the filler cap or pressure-compensation devices. The fuel tank must be situated far enough from the engine to avoid ignition of escaping fuel in case of an accident.

Fuel lines

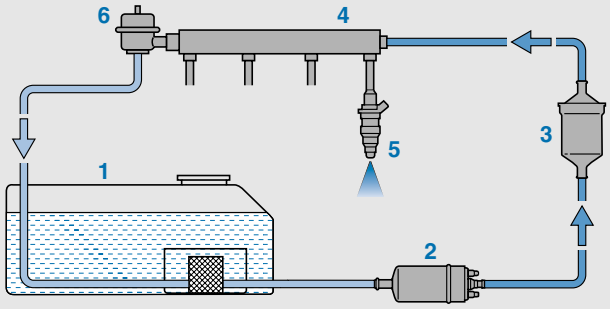
The fuel lines serve to carry the fuel from the fuel tank to the fuel-injection system. Seamless, flexible metal conduit or fuel-resistant hardly combustible material can be used for the fuel lines. These must be routed so that mechanical damage is avoided, and fuel which has evaporated or dripped as a result of malfunctions cannot accumulate or ignite. All fuel-carrying components must be protected against heat that could interfere with correct performance. Gravity feed must not be used in the fuel-supply circuit.

1 Development of fuel-supply systems (examples)

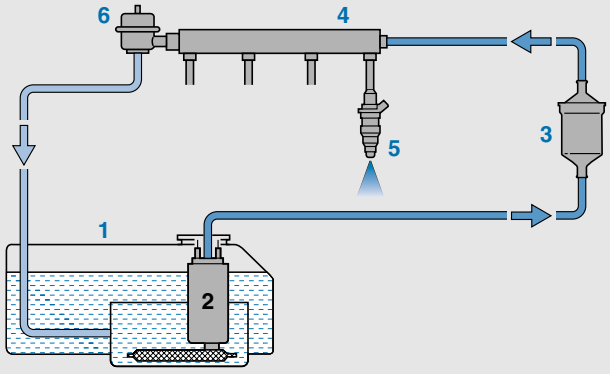
a K-/KE-Jetronic with electric (in-line) fuel pump.



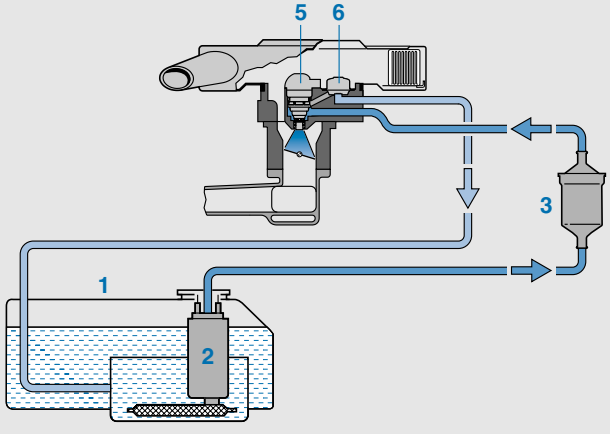
b L-Jetronic/Motronic with electric (in-line) fuel pump.



c L-Jetronic/Motronic with electric (in-tank) fuel pump.



d Mono-Jetronic with electric (in-tank) fuel pump.



UMK1780E

Figure 1
1 Fuel tank
2 Electric fuel pump (EKP)
3 Fuel filter
4 Fuel rail
4a Fuel distributor (K-/KE-Jetronic)
5 Injector
6 Pressure regulator
7 Fuel accumulator (K-/KE-Jetronic)

Manifold fuel injection

Manifold-injection engines generate the A/F mixture in the intake manifold and not in the combustion chamber. Since they were introduced to the market, these engines and their control systems have been vastly improved. Their superior fuel-metering characteristics have enabled them to completely supersede the carburetor engine which also operates with external A/F-mixture formation.

Overview

Regarding smooth running and exhaust-gas behaviour very high demands are made on modern-day vehicles which correspond to the latest state-of-the-art. This leads to strict requirements with respect to the composition of the A/F mixture. Apart from the precision metering of the injected fuel mass as a function of the air drawn in by the engine, it is also imperative that injection of the fuel takes place at exactly the right instant in time.

As a direct result of increasingly severe emission-control legislation, these technical stipulations are being increasingly tightened so that fuel-injection system development is forced to keep pace.

In the manifold-injection field, the electronically controlled multipoint fuel-injection system represents the state-of-the-art. This system injects the fuel intermittently, and individually, for each cylinder directly onto its intake valve(s) (Fig. 1).

Mechanically controlled continuous-injection multipoint systems no longer have any significance for new developments in this field, nor do the single-point (TBI) systems which inject intermittently through a single injector into the intake manifold upstream of the throttle valve.

1 Manifold injection

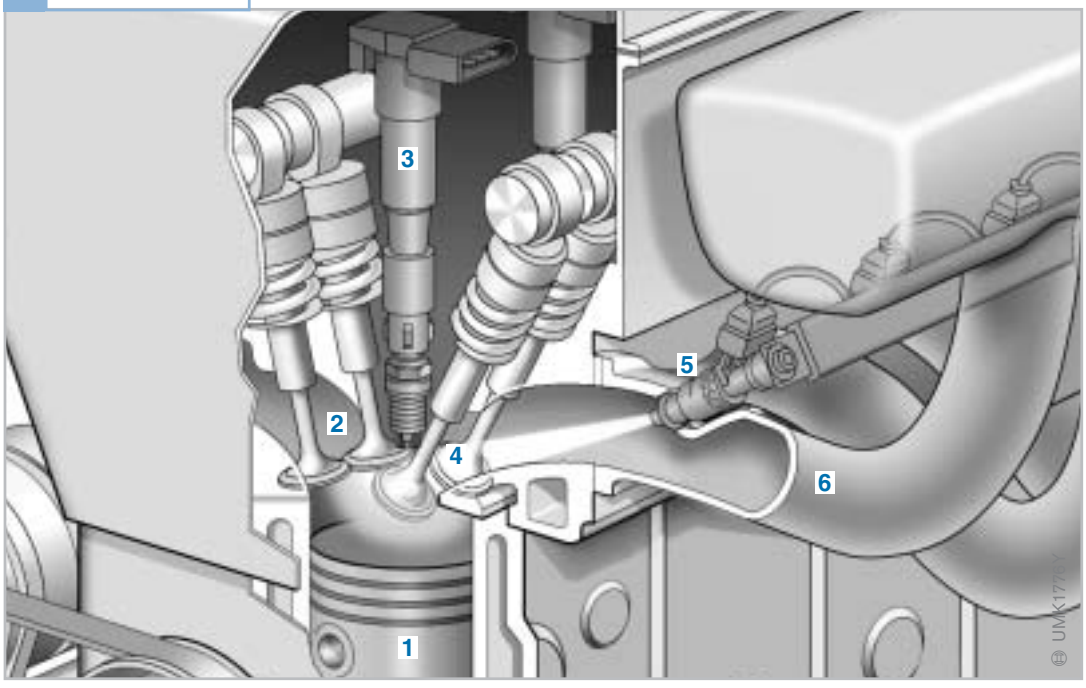


Figure 1
1 Cylinder with piston
2 Exhaust valves
3 Ignition coil with spark plug
4 Intake valves
5 Injector
6 Intake manifold

Robert Bosch GmbH

Operating concept

Gasoline injection systems of the manifold-injection type are characterized by the fact that they generate the A/F mixture outside the combustion chamber, in other words, in the intake manifold (Fig. 1), see "External A/F-mixture formation". The injector (5) sprays the fuel directly onto the intake valves (4) where together with the intake air it forms the A/F mixture which is then drawn into the cylinder (1) past the open intake valves during the subsequent induction stroke. One, two, or even three, intake valves can be used per cylinder.

The intake valves are designed so that the engine's fuel requirements are covered irrespective of operating conditions – at full load and at high engine revs.

A/F-mixture formation

Fuel injection

The electric fuel pump delivers the fuel to the injectors where it is then available for injection at system pressure. Each cylinder is allocated its own injector which injects intermittently into the intake manifold directly onto the intake valve (6). Here the finely atomized fuel evaporates to a great extent, and together with the intake air entering via the throttle plate generates the A/F mixture. In order that enough time is available for the generation of the A/F mixture, the fuel is best sprayed onto the closed intake valve and "stored" there.

Some of the fuel is deposited as a film on the manifold walls in the vicinity of the intake valves. The thickness of the film is a function of the manifold pressure and, therefore, of engine load. For good dynamic engine response, the fuel mass in the wall film must be kept to a minimum. This is achieved by appropriate manifold design and fuel-spray geometry. Since the injector is situated directly opposite the intake valve, the wall-film effects with multipoint injection systems are far less serious than they were with the former TBI and carburetor systems.

Provided the A/F mixture is stoichiometric ($\lambda = 1$), the pollutants generated during the combustion process can to a great extent be removed using the three-way catalytic converter. At the majority of their operating points, manifold-injection engines are therefore operated with this A/F mixture composition.

Measuring the air mass

In order that the A/F mixture can be precisely adjusted, it is imperative that the mass of the air which is used for combustion can be measured exactly. The air-mass meter is situated upstream of the throttle valve. It measures the air-mass flow entering the intake manifold and sends a corresponding electric signal to the engine ECU. As an alternative, there are also systems on the market which use a pressure sensor to measure the intake-manifold pressure. Together with the throttle-valve setting and the engine speed, this data is then used to calculate the intake-air mass. The ECU then applies the data on intake air mass and the engine's instantaneous operating mode to calculate the required fuel mass.

Injection duration

A given length of time is needed for the injection of the calculated fuel mass. This is termed the injection duration, and is a function of the injector's opening cross section and the difference between the intake-manifold pressure and the pressure prevailing in the fuel-supply system.

Robert Bosch GmbH

Electromagnetic fuel injectors

Assignment

The electromagnetic (solenoid-controlled) fuel injectors spray the fuel into the intake manifold at system pressure. They permit the precise metering of the quantity of fuel required by the engine. They are triggered via ECU driver stages with the signal calculated by the engine-management system.

Design and operating concept

Essentially. the electromagnetic injectors (Fig. 1) are comprised of the following components:

- The injector housing (9) with electrical (8) and hydraulic (1) connections,
- The coil for the electromagnet (4),
- The movable valve needle (6) with solenoid armature and sealing ball,
- The valve seat (10) with the injection-orifice plate (7), and the
- Spring (5).

In order to ensure trouble-free operation, stainless steel is used for the parts of the injector which come into contact with fuel. The injector is protected against contamination by a filter strainer (3) at the fuel input.

Connections

On the injectors presently in use, fuel supply to the injector is in the axial direction, that is, from top to bottom ("top feed"). The fuel line is fastened to the injector using a special clamp. Retaining clips ensure reliable alignment and fastening. The seal ring (2) on the hydraulic connection (1) seals off the injector at the fuel rail.

The injector is connected electrically to the engine ECU.

Injector operation

With no voltage across the solenoid (solenoid de-energised), the valve needle and sealing ball are pressed against the cone-shaped valve seat by the spring and the force exerted by the fuel pressure. The fuel-supply system is thus sealed off from the manifold. As soon as the solenoid is energised (excitation current), this generates a magnetic field which pulls in the valve-needle armature. The sealing ball lifts off the valve seat and the fuel is injected. As soon as the excitation current is switched off, the valve needle closes again due to spring force.

Fuel outlet

The fuel is atomized by means of an injection-orifice plate in which there are a number of holes. These holes (spray orifices) are stamped out of the plate and ensure that the injected fuel quantity remains highly reproducible. The injection-orifice plate is insensitive to fuel deposits. The spray pattern of the fuel leaving the injector is a function of the number of orifices and their configuration.

The highly efficient injector sealing at the valve seat is due to the cone/ball sealing principle.

The injector is inserted into the opening provided for it in the intake manifold. The bottom seal ring provides the seal between the injector and the intake manifold. Essentially, the injected fuel quantity per unit of time is determined by

- The fuel-supply system pressure,
- The counterpressure in the intake manifold,
- The geometry of the fuel-exit area.

Types of construction

In the course of time, the injectors have been further and further developed to match them to the ever-increasing demands regarding engineering, quality, reliability, and weight. This has led to a variety of different injector designs.

EV6 injector

The EV6 injector is the standard injector for today's modern fuel-injection systems (Figs. 1 and 2a). It is characterized by its small external dimensions and its low weight. This injector therefore already provides one of the prerequisites for the design of compact intake modules.

In addition, the EV6 is also outstanding with regard to its hot-fuel behaviour, that is, there is very little tendency for vapor-bubble formation when using hot fuel. This facilitates the use of RLFS fuel-supply systems in which the fuel temperature in the injector is higher than with systems featuring fuel return. Thanks to wear-resisting surfaces, the fuel quantities injected by the EV6 remain highly reproducible over long periods of time, and the injector features a long useful life.

Thanks to their highly efficient sealing, these injectors fulfill all future requirements regarding zero evaporation. That is, no fuel vapor escapes from them.

The EV6 variant with "air shrouding" was developed especially to comply with requirements for even better fuel atomization. Finely vaporized fuel can be generated using other methods: In future, in addition to 4-hole injection-orifice plates, multi-orifice plates with between 10 and 12 holes will be used. Injectors equipped with these multi-orifice plates generate a very fine fuel fog.

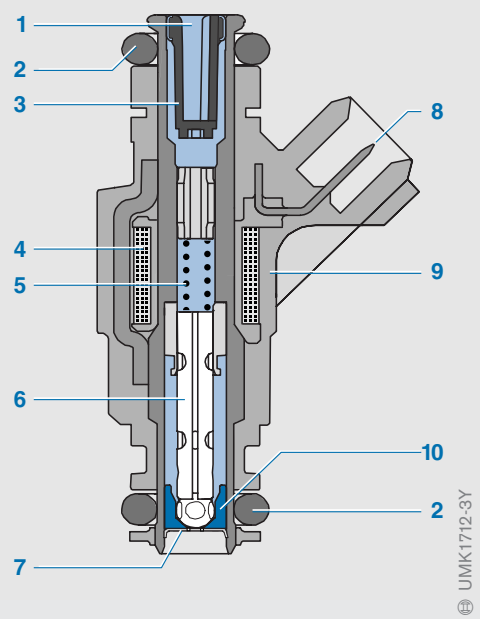
There are a wide variety of injectors available for different areas of application. These feature different lengths, flow classes, and electrical characteristics. The EV6 is also suitable for use with fuels having an ethanol content of as much as 85 %.

EV14 injector

Further injector development has led to the EV14 (Fig. 2b) which is based on the EV6. It is even more compact, a fact which facilitates its integration in the fuel rail.

The EV14 is available in 3 different lengths (compact, standard, long). This makes it possible to adapt individually to the engine's intake-manifold geometry.

1 Design of the EV6 electromagnetic injector



2 Injector versions

Figure 1
1 Hydraulic connection
2 Seal rings (O-rings)
3 Filter strainer
4 Coil
5 Spring
6 Valve needle with armature and sealing ball
7 Injection-orifice plate
8 Electrical connection
9 Injector housing
10 Valve seat

Figure 2
a EV6 Standard
b EV14 Compact

Robert Bosch GmbH

Spray formation

The injector's spray formation, that is, its fuel-spray shape, spray-dispersal angle, and fuel-droplet size, have a considerable influence upon the generation of the A/F mixture. Different versions of spray formation are required in order to comply with the demands of individual intake-manifold and cylinder-head geometries. Fig. 3 shows the most important fuel-spray shapes.

"Pencil" spray

A thin, concentrated, and highly-pulsed fuel spray results from using a single-hole injection-orifice plate. This form of spray practically eliminates the wetting of the manifold wall. Such injectors are most suitable for use with narrow intake manifolds, and in installations in which the fuel has to travel long distances between the point of injection and the intake valve.

The pencil-spray injector is only used in isolated cases due to its low level of fuel atomization.

3 Fuel-spray shapes

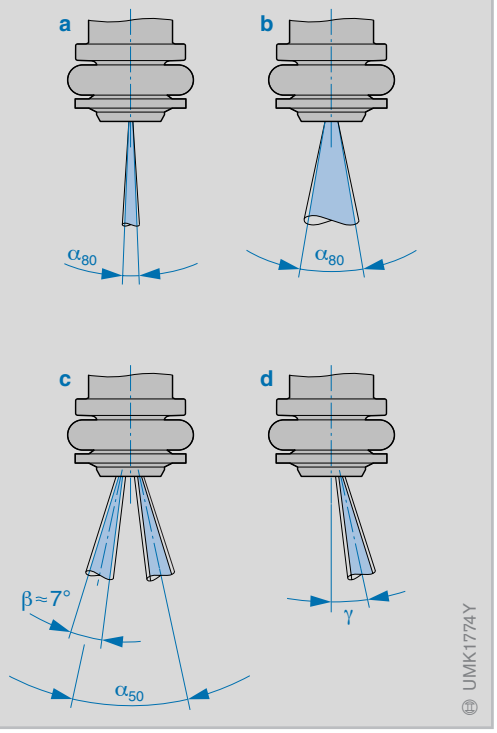


Figure 3
a Pencil spray
b Tapered spray
c Dual spray
d Spray offset angle

α_{80}: 80 % of the injected fuel is within the angle defined by α
α_{50}: 50 % of the injected fuel is within the angle defined by α
β: 70 % of the injected fuel in a single spray is within the angle defined by β
γ: Spray offset angle

Tapered spray

A number of individual jets of fuel leave the injection-orifice plate. The tapered spray cone results from the combination of these fuel jets.

Although engines with only 1 intake valve per cylinder typically use tapered-spray injectors, they are also suitable for engines with 2 intake valves per cylinder.

Dual spray

The dual-spray formation principle is often applied on engines with 2 intake valves per cylinder. Engines with 3 intake valves per cylinder must be equipped with dual-spray injectors.

The holes in the injection-orifice plate are so arranged that two fuel sprays leave the injector and impact against the respective intake valve or against the web between the intake valves. Each of these sprays can be formed from a number of individual sprays (2 tapered sprays).

The spray offset angle

Referred to the injector's principle axis, the fuel spray in this case (single spray and dual spray) is at an angle, the spray offset angle (γ). Injectors with this spray shape are mostly used when installation conditions are difficult.

Types of fuel injection

In addition to the duration of injection, a further parameter which is important for optimisation of the fuel-consumption and exhaust-gas figures is the instant of injection referred to the crankshaft angle. Here, the possible variations are dependent upon the type of injection actually used (Fig. 1).

The new injection systems provide for either sequential fuel injection or cylinder-individual fuel injection (SEFI and CIFI respectively).

Robert Bosch GmbH

Simultaneous fuel injection

All injectors open and close together in this form of fuel injection. This means that the time which is available for fuel evaporation is different for each cylinder. In order to nevertheless obtain efficient A/F-mixture formation, the fuel quantity needed for the combustion is injected in two portions. Half in one revolution of the crankshaft and the remainder in the next. In this form of injection, the fuel for some of the cylinders is not stored in front of the particular intake valve but rather, since the valve has opened, the fuel is injected into the open intake port. The start of injection cannot be varied.

Group injection

Here, the injectors are combined to form two groups. For one revolution of the crankshaft, one injector group injects the total fuel quantity required for its cylinders, and for the next revolution the second group injects.

This configuration enables the start of injection to be selected as a function of engine-operating point. Apart from this, the undesirable injection into open inlet ports is avoided. Here too, the time available for the evaporation of fuel is different for each cylinder.

Sequential fuel injection (SEFI)

The fuel is injected individually for each cylinder, the injectors being actuated one after the other in the same order as the firing sequence. Referred to piston TDC, the duration of injection and the start of injection are identical for all cylinders, and the fuel is stored in front of each cylinder.

Start of injection is freely programmable and can be adapted to the engine's operating state.

Cylinder-individual fuel injection (CIFI)

This form of injection provides for the greatest degree of design freedom. Compared to sequential fuel injection, CIFI has the advantage that the duration of injection can be individually varied for each cylinder. This permits compensation of irregularites, for instance with respect to cylinder charge.

1 Manifold fuel injection: Types of fuel injection

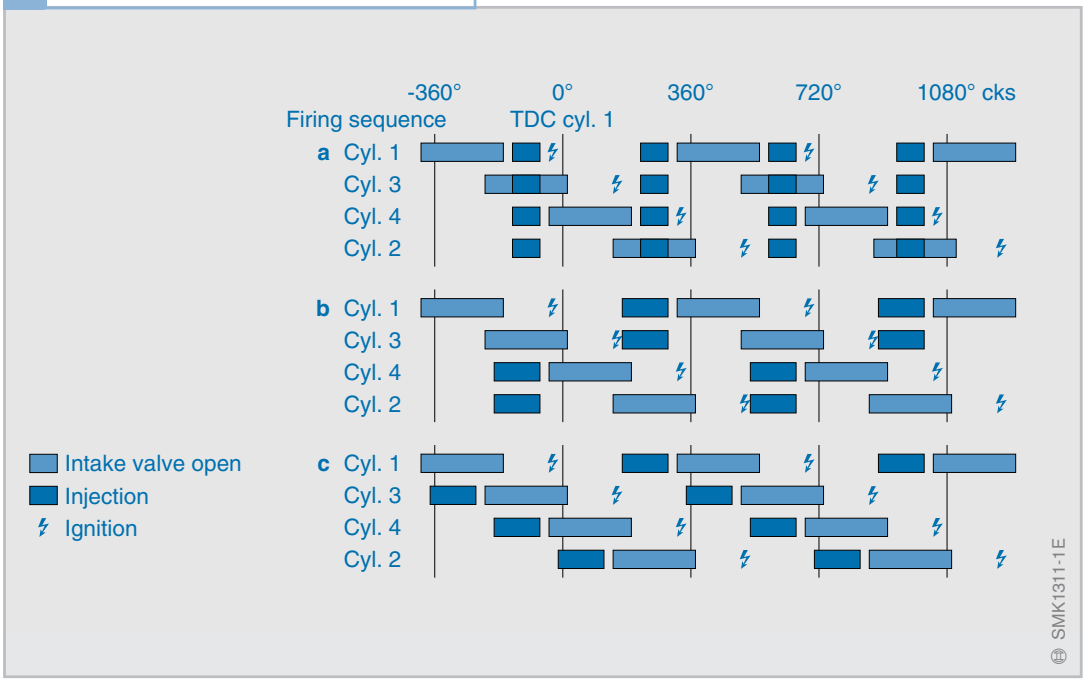


Figure 1
a Simultaneous fuel injection
b Group fuel injection
c Sequential fuel injection (SEFI) and cylinder-individual fuel injection (CIFI)

Gasoline direct injection

Gasoline direct-injection engines generate the A/F mixture in the combustion chamber. During the induction stroke, only the combustion air flows past the open intake valve and into the cylinder. The fuel is injected directly into the cylinders by special injectors.

Overview

The demand for higher-power engines, coupled with the requirement for reduced fuel consumption, were behind the "re-discovery" of gasoline direct injection. As far back as 1937, an engine with mechanical gasoline direct injection took to the air in an airplane. In 1952, the "Gutbrod" was the first passenger car with a series-production mechanical gasoline direct-injection engine, and in 1954 the "Mercedes 300 SL" followed.

At that time, designing and building a direct-injection engine was a very complicated business. Moreover, this technology made extreme demands on the materials used. The engine's service life was a further problem.

These facts all contributed to it taking so long for gasoline direct injection to achieve its breakthrough.

1 Gasoline direct injection: Components

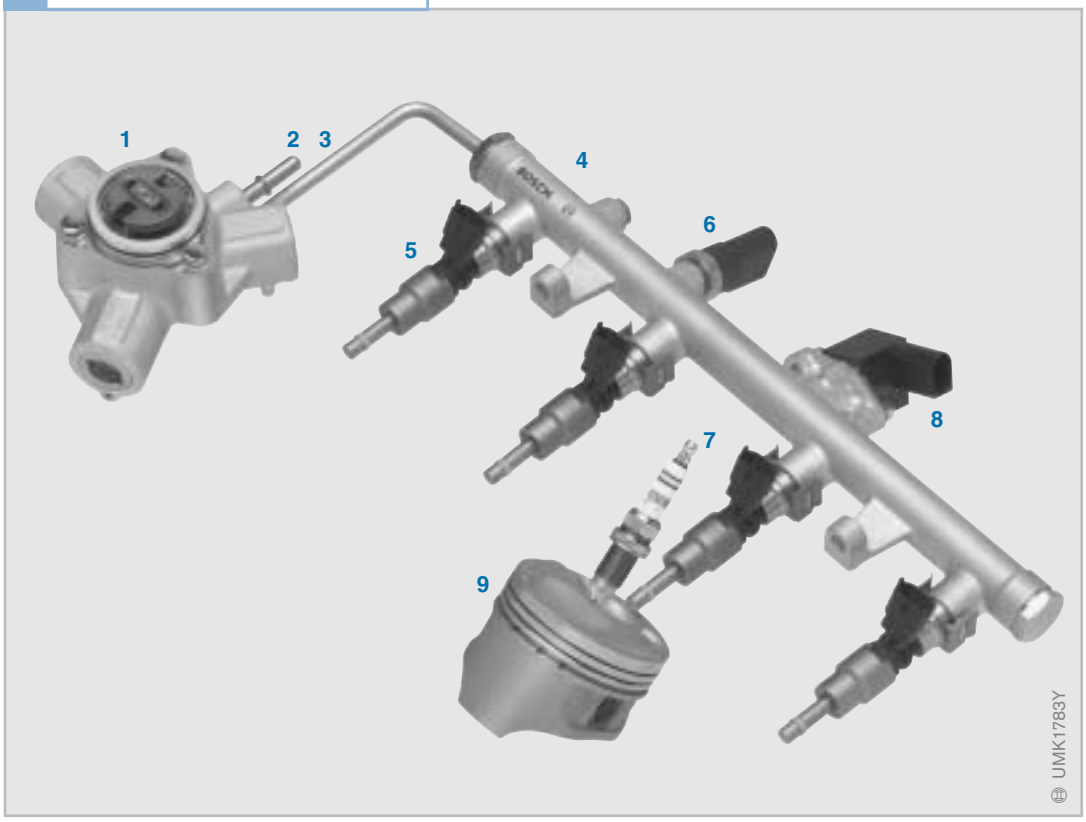


Figure 1
1 High-pressure pump
2 Low-pressure connection
3 High-pressure line
4 Fuel rail
5 High-pressure injectors
6 High-pressure sensor
7 Spark plug
8 Pressure-control valve
9 Piston

Operating concept

Gasoline direct-injection systems are characterized by injecting the fuel directly into the combustion chamber at high pressure. Similar to the diesel engine, A/F-mixture formation takes place inside the cylinder (internal A/F-mixture formation).

Generation of high-pressure

The electric fuel pump delivers fuel to the high-pressure pump (Fig. 1, Pos. 1) at a primary pressure of 0.3...0.5 MPa (3...5 bar). Depending on the engine operating point (required torque and engine speed), the high-pressure pump then generates the system pressure which forces the fuel, which is now at high pressure, into the rail (4) where it is stored until required for injection.

The fuel pressure is measured by the high-pressure sensor (6) and adjusted to values between 5...12 MPa by the pressure-control valve (8).

The high-pressure injectors (5) are installed in the rail (also referred to as the "Common rail") and, when triggered by the engine ECU, inject the fuel directly into the combustion chambers.

A/F-mixture formation

The injected fuel is finely atomized due to the very high injection pressure. Together with the drawn-in air, it forms the A/F mixture in the combustion chamber. Depending upon the engine's operating mode, the fuel is injected in such a manner that an A/F mixture with $\lambda \leq 1$ is evenly distributed throughout the complete combustion chamber (homogeneous operation), or a stratified-charge A/F-mixture cloud ($\lambda \leq 1$) is formed around the spark plug (lean-burn operation or stratified-charge operation). During stratified-charge operation, the remainder of the cylinder is filled with either freshly drawn-in air, with inert gas returned to the cylinder by EGR, or with a very lean A/F mixture. The overall A/F mixture then has ltotal $\lambda_{total} > 1$.

The various methods of running the engine as listed above are referred to as the engine's operating modes. On the one hand, the selection of the operating mode to be applied is a function of engine speed and desired torque, and on the other it depends upon functional requirements such as the regeneration of the accumulator-type catalytic converter.

Torque

During stratified-charge operation, the injected fuel mass is decisive for the generated torque. The excess air permits "unthrottled" operation, also at part load, with the throttle opened wide. This measure reduces the pumping (exhaust and refill) work, and therefore also serves to lower the fuel consumption.

In homogeneous and lean-burn operation at $\lambda > 1$ and homogeneous A/F-mixture distribution, "unthrottling" also results in fuel savings, although not to the same extent as in stratified-charge operation.

In homogeneous operation at $\lambda \leq 1$, the gasoline direct-injection engine for the most part behaves the same as a manifold-injection engine.

Exhaust treatment

The catalytic converter is responsible for removing the pollutants from the exhaust gas. In order to operate with maximum efficiency, the 3-way catalytic converter needs a stoichiometric A/F mixture. Due to excess air, lean-burn operation results in increased levels of NO_x emissions which are stored temporarily in an accumulator-type NO_x catalytic converter. These are then reduced to nitrogen, carbon dioxide and water by running the engine briefly with excess air.

Rail

The rail stores the fuel delivered by the high-pressure pump and distributes it to the high-pressure fuel injectors. The rail's volume is sufficient to compensate for pressure pulsations in the fuel circuit.

An aluminum rail is used. Design and construction (volume, dimensions, weight etc.) are specific to the engine and the system.

The rail is provided with connections for a number of the injection-system components (high-pressure pump, pressure-control valve, high-pressure sensor, high-pressure injectors). Construction guarantees that there are no leaks in the rail itself, nor at its interfaces.

High-pressure pumps

Assignment

It is the job of the high-pressure pump (HDP) to compress the fuel delivered by the electric fuel pump at a primary pressure of 0.3...0.5 MPa. It must provide enough fuel at the pressure (5...12 MPa) needed for the high-pressure injection.

Initially, when starting the engine, the fuel is injected at the primary pressure. The high pressure is built up when the engine runs up to speed. The minimal level of pumping-flow pulsation means so that there is very little pulsation in the rail.

In order to prevent the fuel mixing with an oil lubricant, the high-pressure pump is cooled and lubricated by fuel.

1 Three-cylinder high-pressure pump HDP1 (axial section)

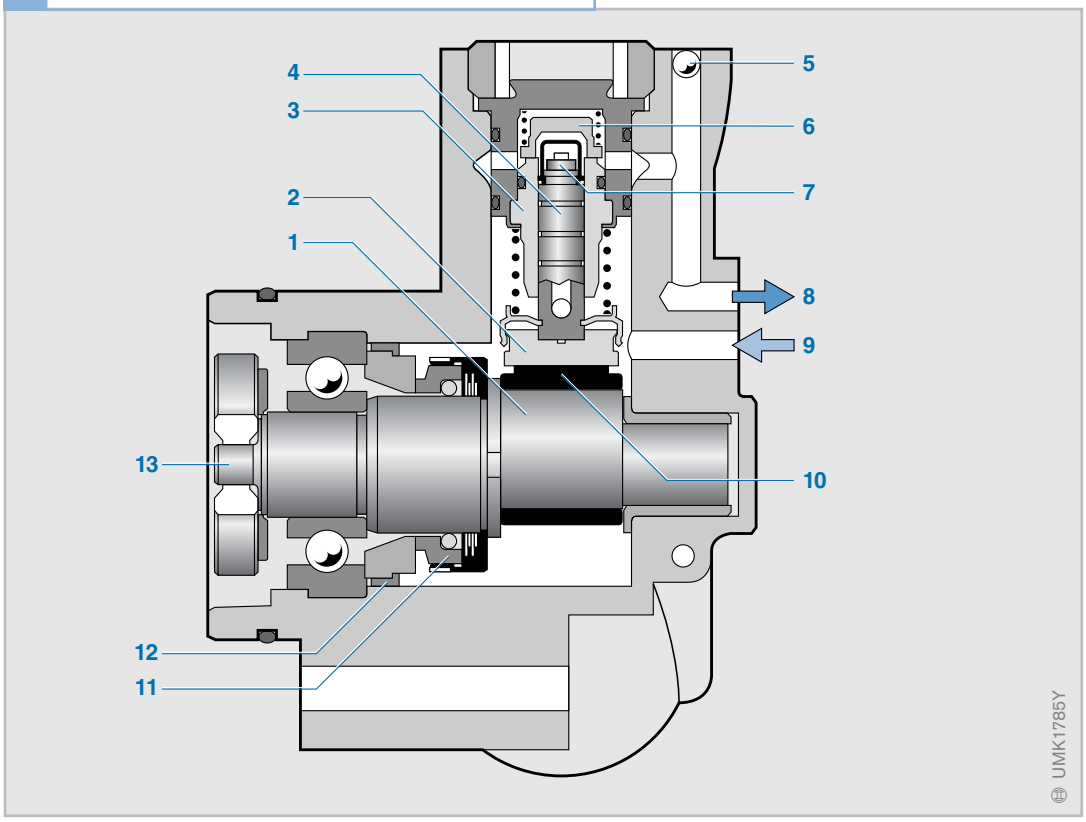


Figure 1

1 Eccenter cam
2 Sliding block
3 Pumping element with pump piston (hollow piston, fuel inlet)
4 Sealing ball
5 Outlet valve
6 Inlet valve
7 High-pressure con nection to the rail
8 Fuel inlet (low presure)
9 Eccenter ring
10 Axial face seal
11 Static seal
12 Driveshaft

2 Three-cylinder high-pressure pump HDP 1 (cross-section)

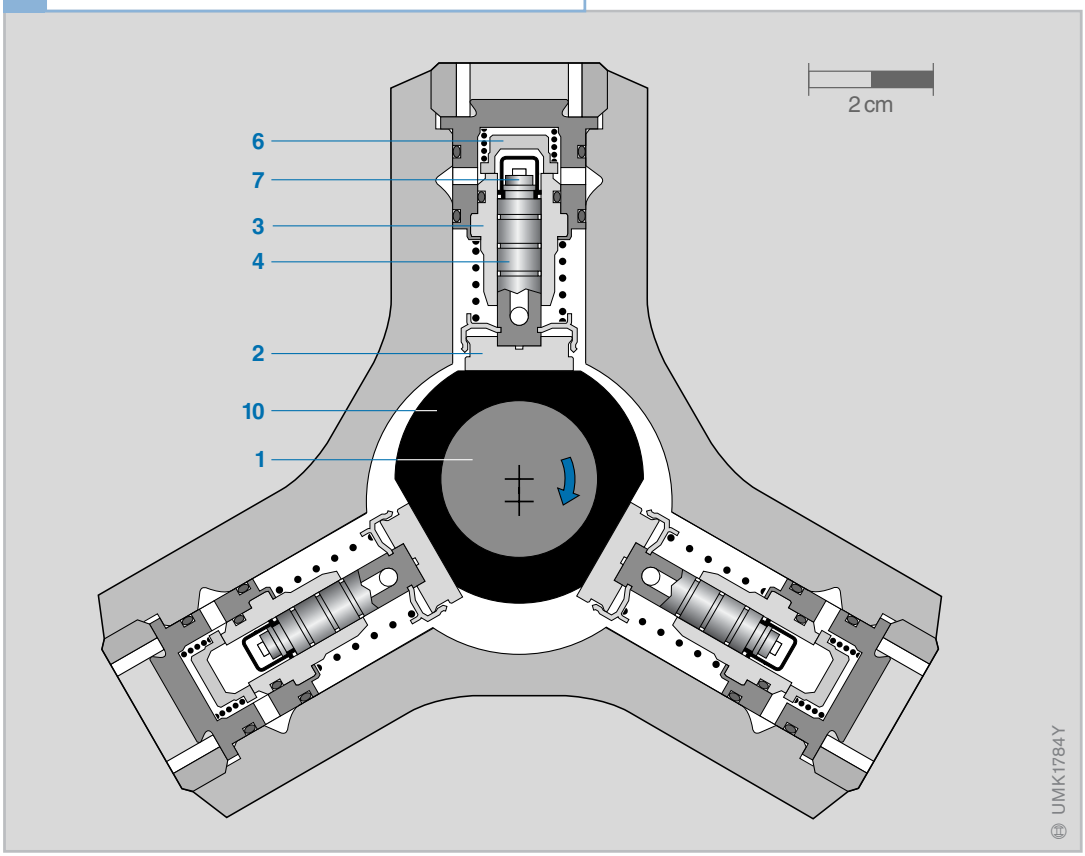


Figure 2
(Position numbers identical to Fig. 1)
1 Eccenter cam
2 Sliding block
3 Pumping element with pump piston
5 Outlet valve
6 Inlet valve
9 Eccenter ring

Three-cylinder high-pressure pump HDP1

There are many different types of high-pressure pumps available. Fig. 1 shows an axial section, and Fig. 2 a cross section through the HDP1 three-cylinder radial-piston pump. Driven by the engine camshaft, the driveshaft (12) rotates with the eccenter cam (1) which is responsible for the up and down motion of the pistons (3) in their cylinders. When the piston moves downward, fuel flows at the primary pressure of 0.3...0.5 MPa from the fuel line through the hollow pump piston and the inlet valve (6) into the pump cylinder. When the piston moves upward this volume of fuel is compressed and when the rail pressure is reached the outlet valve (5) opens and the fuel is forced out to the high-pressure connection (7).

The use of three pump cylinders at an angle of 120° to each other results in very low levels of residual pulsation in the rail. The delivery quantity is proportional to the rotational speed. So that there is always enough fuel available, and in order to limit the fuel warm-up in the rail, at maximum delivery the high-pressure pump delivers slightly more fuel than the maximum needed by the engine. The pressure-control valve releases the pressure of the excess fuel and then directs this into the return line.

Single-cylinder high-pressure pump HDP2

The HDP2 single-cylinder pump is a cam-driven radial-piston pump with variable delivery quantity. When the piston moves downward, fuel flows at the primary pressure of 0.3...0.5 MPa from the fuel line, through the inlet valve and into the pump cylinder. When the piston moves upwards this volume

of fuel is compressed and forced into the rail as soon as it exceeds the rail pressure.

The pump chamber and the fuel inlet are connected with each other through a triggerable delivery-quantity control valve. If this valve is triggered and opens before the end of the delivery stoke, the pressure in the pump chamber collapses and the fuel flows back into the fuel inlet. This means that the delivery-control valve has the same function as the pressure-control valve in the three-cylinder HDP1 pump.

In order to adjust the delivery quantity, the quantity control valve remains closed from pump-cam BDC until a given stroke has been completed. Once the required rail pressure is reached the valve opens and prevents further pressure increase in the rail.

The maximum delivery quantity (*l*/h) is a function of the rotational speed, the number of cams, and the cam lift. The delivery quantity can be adjusted to comply with requirements by triggering the control valve accordingly.

The non-return valve between the pump chamber and the rail prevents the rail pressure dropping when the delivery-quantity control valve opens.

1 Section through the pressure-control valve

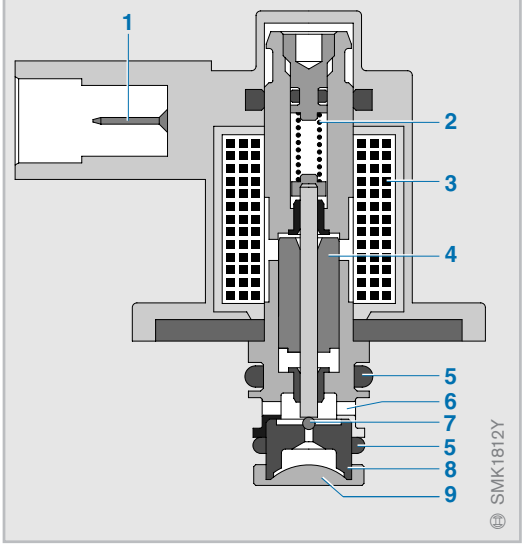


Figure 1
1 Electrical connection
2 Spring
3 Solenoid coil
4 Solenoid armature
5 Seal rings (O-rings)
6 Outlet passage
7 Valve ball
8 Valve seat
9 Inlet with inlet strainer

Pressure-control valve

Assignment

The pressure-control valve is located between the rail and the low-pressure side of the HDP1 high-pressure pump. It adjusts the required pressure in the rail by changing the flow cross-section. The excess fuel delivered by the HDP1 flows into the low-pressure circuit.

Design and operating concept

The solenoid is triggered by a pwm signal (Fig. 1, Pos. 3). The valve ball (7) lifts from the valve seat (8) and in doing so changes the valve's flow cross-section as required.

With no current flowing, the pressure-control valve is closed. This is a safety measure to ensure adequate rail pressure in case of malfunction in the electrical-triggering circuit. A pressure-limiting function is incorporated to prevent excessive rail pressure which could otherwise damage the components.

Robert Bosch GmbH

Rail-pressure sensors

Assignment

The rail-pressure sensors used in the Common Rail and MED-Motronic systems measure the fuel pressure in the high-pressure fuel reservoir (fuel rail). Precisely maintaining the stipulated fuel pressure in the rail is of extreme importance with respect to the engine's power output, toxic emissions, and noise. Fuel pressure is controlled in a special control loop, deviations from desired value being compensated for by an open-loop or closed-loop pressure-control valve.

Very tight tolerances apply to the rail-pressure sensors, and in the main operating range, the measuring accuracy is below 2% of the measuring range.

Rail-pressure sensors are used with the following engine systems:

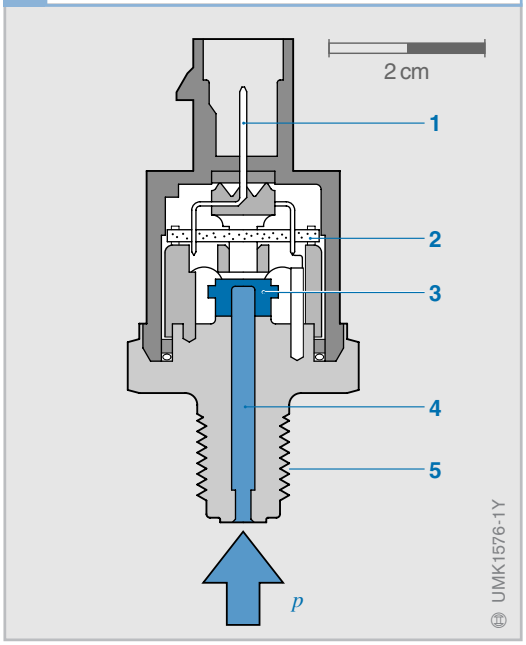
- *Common Rail diesel accumulator-type injection system*
  The maximum working pressure p_{max} (rated pressure) is 160 MPa (1600 bar).

- *Gasoline direct injection MED-Motronic*
  The working pressure in such a gasoline direct injection system is a function of the torque and engine speed.
  It is 5 ... 12 MPa (50 ... 120 bar).

Design and operating concept

A steel diaphragm is at the heart of the rail-pressure sensor. Deformation-dependent measuring resistors are vapor deposited on the diaphragm in the form of a bridge circuit (Fig. 1, Pos. 3). The sensor's measuring range is a function of the diaphragm thickness (thicker diaphragms are used for higher pressures, and thinner ones for lower pressures).
As soon as the pressure to be measured is applied to one side of the diaphragm via the pressure connection (4) the deformation-dependent resistors change their values due to the bending of the diaphragm (approx. 20 µm at 1500 bar). By means of collecting leads, the 0 ... 80 mV output voltage signal generated by the bridge is transferred to an evaluation circuit (2) in the sensor and amplified to 0 ... 5 V. This is then passed on to the ECU which uses it, together with a stored characteristic curve, to calculate the pressure (Fig. 2)

1 Rail-pressure sensor (design)



2 Rail-pressure sensor: Charcteristic curve (example)

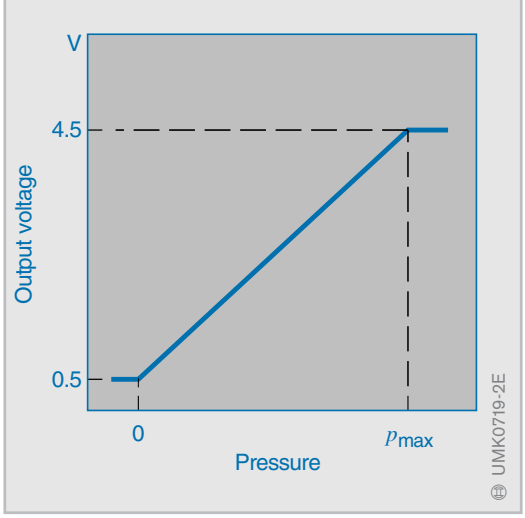


Fig. 1
1 Electrical connection (plug)
2 Evaluation circuit
3 Steel diaphragm with deformation-dependent resistors
4 Pressure connection
5 Mounting thread

High-pressure injector

Assignment

The high-pressure injector represents the interface between the rail and the combustion chamber. Its job is to meter the fuel, and by means of the fuel's atomisation achieve controlled mixing of the fuel and air in a specific area of the combustion chamber. Depending upon the required operating mode, the fuel is either concentrated in the vicinity of the spark plug (stratified charge), or evenly distributed throughout the combustion chamber (homogeneous distribution).

1 High-pressure injector (HDEV): Design

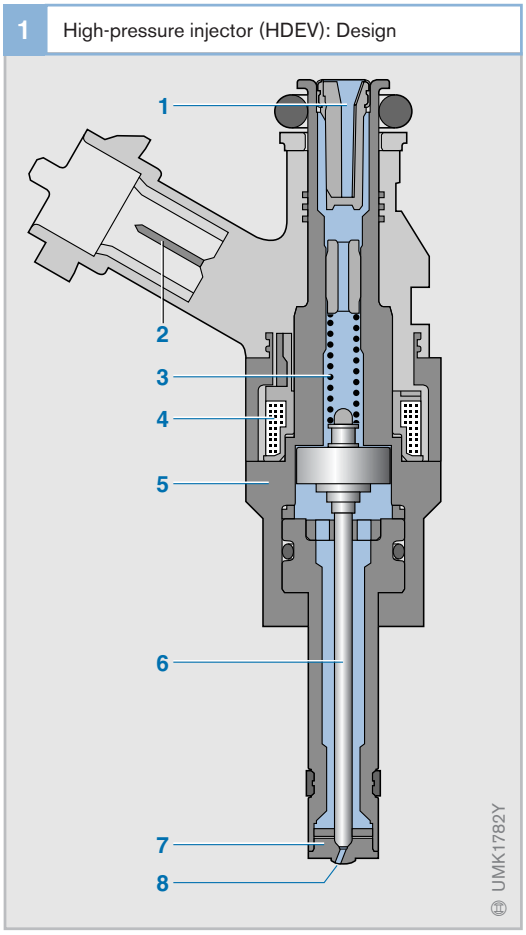


Figure 1
1 Fuel inlet with fine strainer
2 Electrical connections
3 Spring
4 Solenoid
5 Injector housing
6 Nozzle needle with solenoid armature
7 Valve seat
8 Injector outlet passage

Design and operating concept

The high-pressure injector (Fig. 1) comprises the following components:

- Injector housing (5),
- Valve seat (7),
- Nozzle needle with solenoid armature (6),
- Spring (3), and
- Solenoid (4).

A magnetic field is generated when the solenoid coil is energized (current flows). This lifts the valve needle from the valve seat against the force of the spring and opens the injector outlet passage (8). Fuel is then injected into the combustion chamber due to the difference between rail pressure and combustion-chamber pressure.

When the energising current is switched off, the spring forces the needle back down against its seat and injection stops.

The injector opens very quickly, guarantees a constant opened cross-section during the time it is open, and closes against the rail pressure. Taking a given opened cross-section, the injected fuel quantity is therefore dependent upon the rail pressure, the counter-pressure in the combustion chamber, and the length of time the injector remains open. Excellent fuel atomisation is achieved thanks to the special nozzle geometry at the injector tip.

Compared to manifold injection, gasoline direct injection can boast faster injection, improved precision of spray alignment, and better formation of the fuel spray.

Technical requirements

Compared with manifold injection, gasoline direct injection differs mainly in its higher fuel pressure and the far shorter time which is available for directly injecting the fuel into the combustion chamber.

Robert Bosch GmbH

Fig. 2 underlines the technical demands made on the injector. With manifold injection, two revolutions of the crankshaft are available for injecting the fuel into the manifold. At an engine speed of 6,000 min^{-1}, this corresponds to 20 ms.

In the case of gasoline direct injection though, considerably less time must suffice. During homogeneous operation, the fuel must be injected in the induction stroke. In other words, only half a crankshaft rotation is available for the injection process. Referred to the same engine speed as with manifold injection (6,000 min^{-1}), this corresponds to an injection duration of only 5 ms.

For gasoline direct injection, the fuel requirement at idle referred to that at WOT is far lower than with manifold injection (factor 1:12). At idle, this results in an injection duration of approx. 0.4 ms.

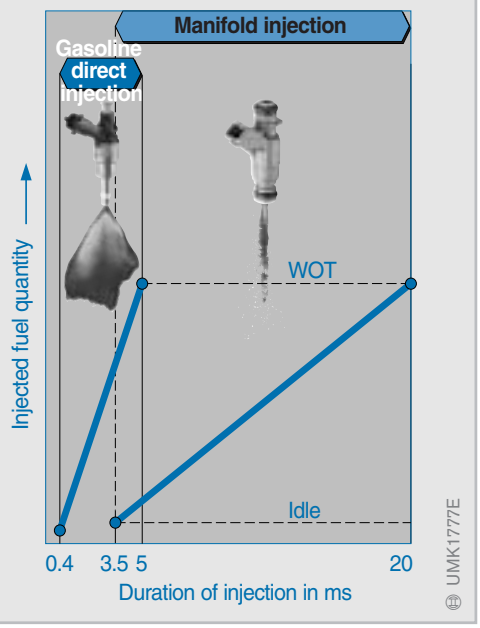
Triggering the HDEV high-pressure injector

The injector must be triggered with a highly complex current characteristic in order to comply with the requirements for defined, reproducible fuel-injection processes (Fig. 3). The initial triggering signal from the microcontroller in the engine ECU is simply a digital signal (a). A special triggering module uses this signal to generate the actual triggering signal (b) with which the HDEV driver stage triggers the injector.

A booster capacitor is used to generate the 50...90 V trigger voltage which is high enough to provide a high current at the start of the switch-on process so that the valve needle can lift off of the valve seat very quickly (c). Once the valve needle has lifted (maximum needle lift), only a very low triggering current suffices to maintain the needle at a constant opened position. With the needle's opened position constant, the injected fuel quantity is proportional to the injection duration (d).

The calculations for the duration of injection take into account the premagnetisation time before the valve needle actually lifts.

2 Comparison between gasoline direct injection and manifold injection



3 Signal characteristic for triggering the HDEV high-pressure injector

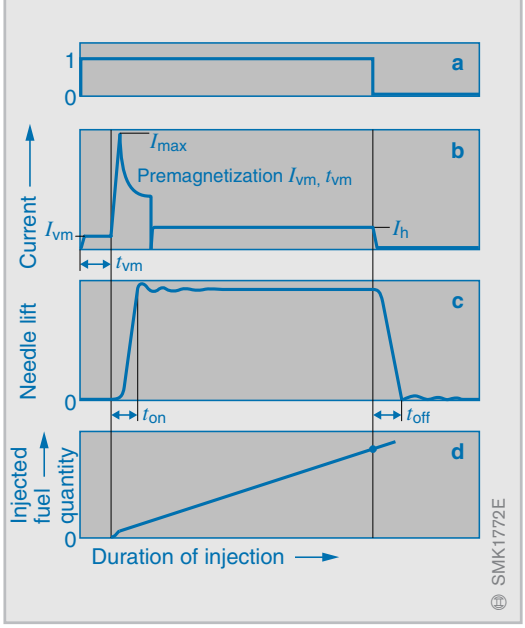


Figure 2
Injected fuel quantity as a function of the duration of injection

Figure 3
a Triggering signal
b Injector current characteristic
c Needle lift
d Injected fuel quantity

Robert Bosch GmbH

Combustion process

The combustion process is defined as the way in which A/F-mixture formation and energy conversion take place in the combustion chamber.

Depending upon the combustion process concerned, flows of air are generated in the combustion chamber. In order to obtain the required charge stratification, the injector injects the fuel into the air flow in such a manner that it evaporates in a defined area. The air flow then transports the A/F-mixture cloud in the direction of the spark plug so that it arrives there at the moment of ignition.

Two basically different combustion processes are possible:

1 Air-flow conditions for the various combustion processes

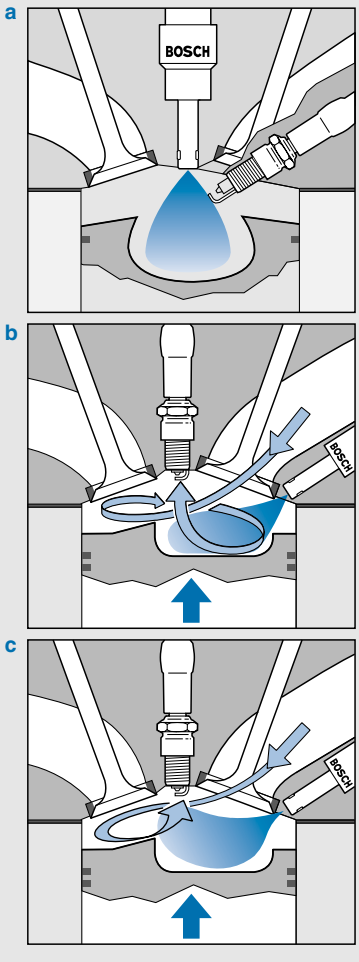


Figure 1
a Spray-guided
b Wall-guided swirl air flow
c Wall-guided tumble air flow

Spray-guided combustion process

The spray-guided process is characterised by the fuel being injected in the spark plug's immediate vicinity where it also evaporates (Fig. 1a). In order to be able to ignite the A/F mixture at the correct moment in time (ignition point), it is imperative that spark plug and injector are exactly positioned, and that the spray direction is precisely aligned.

With this process, the spark plug is subjected to considerable thermal stressing since under certain circumstances the hot spark plug can be directly impacted by the relatively cold jet of injected fuel.

Wall-guided combustion process

In the case of the wall-guided process, one differentiates between two possible flows of air which are the result of specific intake-port and piston design. The injector injects into this air flow which transports the resulting A/F mixture to the spark plug in the form of a closed A/F-mixture cloud.

Swirl air flow

The air drawn by the piston through the open intake valve and into the cylinder generates a turbulent flow (rotational air movement) along the cylinder wall (Fig. 1b). This process is also designated "swirl combustion process".

Tumble air flow

This process produces a cylindrical air flow, or tumbling air flow, which in its movement from top to bottom is deflected by a pronounced piston recess so that it then moves upwards in the direction of the spark plug (Fig. 1c).

Robert Bosch GmbH

A/F-mixture formation

Assignment

It is the job of the A/F-mixture formation to provide a combustible A/F mixture which is to be as homogeneous as possible.

Technical requirements

In the "homogeneous" mode of operation (homogeneous $\lambda \leq 1$ and homogeneous lean-burn), this A/F mixture is distributed homogeneously throughout the whole of the combustion chamber. During stratified-charge operation on the other hand, the A/F mixture is only homogeneous within a restricted area, while the remaining areas of the combustion chamber are filled with inert gas or fresh air.

All fuel must have evaporated before a gas mixture or gas-vapor mixture can be termed homogeneous. A number of factors influence this process:

- Combustion-chamber temperature,
- Fuel-droplet size, and
- The time which is available for fuel evaporation.

Influencing factors

Temperature influence

Depending upon temperature, pressure, and combustion-chamber geometry, an A/F mixture (air/gasoline) is combustible at $\lambda = 0.6...1.6$. Since gasoline cannot evaporate completely at low temperatures, this means that under these conditions more fuel must be injected in order to obtain a combustible A/F mixture.

A/F-mixture formation in the homogeneous operation mode

The fuel is injected as soon as possible so that the maximum length of time is available for formation of the A/F mixture. This is why the fuel is injected in the induction stroke during homogeneous operation. The intake air can then assist in achieving rapid evaporation of the fuel and efficient homogenisation of the A/F mixture.

A/F-mixture formation in the stratified-charge mode

The configuration of the combustible A/F-mixture cloud which is in the vicinity of the spark plug at the instant of ignition is decisive for the stratified-charge mode. This is why the fuel is injected during the compression stroke so that a cloud of A/F mixture is generated which can be transported to the vicinity of the spark plug by the air flows in the combustion chamber, and by the piston as it moves upwards. The ignition point is a function of the engine speed and the required torque.

Penetration depth

The fuel-droplet size in the injected fuel is a function of injection pressure and combustion-chamber pressure. Higher injection pressures result in smaller droplets which then vaporize quicker. Taking a constant combustion-chamber pressure, the so-called penetration depth increases along with increasing injection pressure. The penetration depth is defined as the distance travelled by the individual fuel droplet before it vaporizes completely.

The cylinder wall or the piston will be wetted with fuel if the distance needed for full vaporization exceeds the distance from the injector to the combustion-chamber wall. If the fuel on the cylinder wall and piston has not vaporized before the ignition point, either no combustion takes place at all, or it is incomplete.

Robert Bosch GmbH

Operating modes

There are six operating modes in use with gasoline direct injection (Fig. 1):

- Stratified-charge mode,
- Homogeneous mode,
- Homogeneous and lean-burn,
- Homogeneous and stratified-charge,
- Homogeneous/anti-knock,
- Stratified-charge/cat-heating.

These operating modes permit the best-possible adaptation for each and every engine operating state. During actual driving, the driver does not notice the change-overs between operating modes since these take place without torque surge.

The lines in the diagram (Fig. 1) show which operating modes are passed through when accelerating strongly (pronounced changes in torque with at first unchanged engine speed), and when accelerating gently (slight changes in torque with increasing engine speed).

Stratified-charge mode

In the lower torque range at speeds up to approx. 3000 min^{-1}, the engine is operated in the stratified-charge mode. Here, the injector injects the fuel during the compression stroke shortly before the ignition point. During the brief period before the ignition point the air flow in the combustion chamber transports the A/F mixture to the spark plug. Due to the late injection point, there is not sufficient time to distribute the A/F mixture throughout the complete combustion chamber.

Referred to the combustion chamber as a whole, the A/F mixture is very lean in the stratified-charge mode. The untreated NO_x emissions are very high when the excess-air level is high. In this operating mode, the best remedy is to use a high EGR rate, whereby the recirculated exhaust gas reduces the combustion temperature and, as a result, lowers the temperature-dependent NO_x emissions.

The parameters "Engine speed" and "Torque" define the limits for stratified-charge operation. In the case of excessive torque, soot is generated due to zones of local rich-mixture. If engine speed is too high, charge stratification and efficient transport of the A/F mixture to the spark plug can no longer be maintained due to excessive turbulence.

Homogeneous mode

For high torques and high engine speeds the engine is operated in the homogeneous mode $\lambda = 1$ (in exceptional cases with $\lambda < 1$) instead of in the stratified-charge mode. Injection starts in the induction stroke, so that there is sufficient time for the A/F mixture to be distributed throughout the whole of the combustion chamber. The injected fuel mass is such that the A/F mixture ratio is stoichiometric or, in exceptional cases, has slightly excessive fuel ($\lambda \leq 1$).

Since the whole of the combustion chamber is utilised, the homogeneous mode is required when high levels of torque are demanded. In this operating mode, emissions of untreated exhaust gas are also low due to the stoichiometric A/F mixture.

In homogeneous operation, combustion to a great extent corresponds to the combustion for manifold injection.

1 Operating-mode characteristic curves for gasoline direct injection

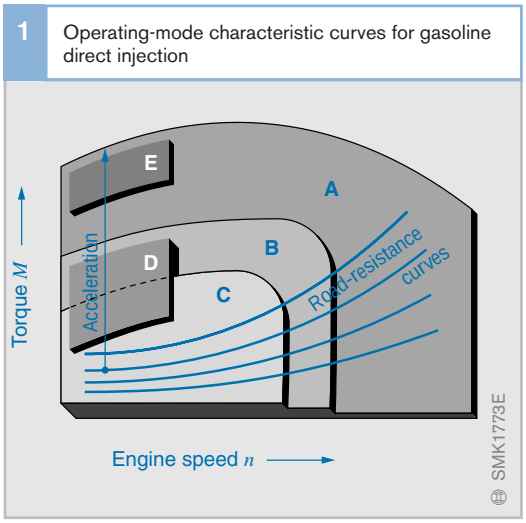


Figure 1
A Homogeneous operation with $\lambda = 1$, this operating mode is possible in all operating ranges
B Lean-burn or homogeneous operation, $\lambda = 1$ with EGR; this operating mode is possible in area C and area D
C Stratified-charge operation with EGR

Operating modes with dual injection:
C Stratified-charge/cat-heating mode, same area as stratified-charge operation with EGR
D Homogeneous and stratified-charge
E Homogeneous/anti-knock

Robert Bosch GmbH

Homogeneous and lean-burn mode

In the transitional range between stratified-charge and homogeneous mode, the engine can be run with a homogeneous lean A/F mixture (λ>1). Since the pumping losses are lower due to "non-throttling", in the homogeneous and lean-burn mode, fuel consumption is lower than in the homogeneous mode with $\lambda \leq 1$.

Homogeneous and stratified-charge mode

The complete combustion chamber is filled with a homogeneous lean A/F mixture. This mixture is generated by injecting a small quantity of fuel during the induction stroke.

Fuel is injected a second time (dual injection) during the compression stroke. This leads to a richer zone forming in the area of the spark plug. This stratified charge is easily ignitable and then ignites the rest of the homogeneous mixture in the remainder of the combustion chamber.

The homogeneous and stratified-charge mode is activated for a number of cycles during the transition between stratified-charge and homogeneous mode. This enables the engine management system to better adjust the torque during the transition. Due to the conversion to energy of the very lean A/F mixture $\lambda > 2$, the NO_x emissions are also reduced.

The distribution factor between each injection is approx. 75 %. That is, 75 % of the fuel is injected in the first injection which is responsible for the homogeneous basic A/F mixture.

Steady-state operation using dual injection at low engine speeds in the transitional range between stratified-charge and homogeneous mode reduces the soot emissions compared to stratified-charge operation, as well as lowering fuel consumption compared to homogeneous operation.

Homogeneous/anti-knock mode

In this operating mode, since the charge stratification hinders knock, the use of dual injection at WOT, together with ignition-angle shift in the retard direction as needed to avoid "knock", can be dispensed with. At the same time, the favorable ignition point also leads to higher torque.

Stratified-charge/cat-heating

Another form of dual injection makes it possible to quickly heat up the exhaust-gas system, although this must be optimized before this solution can be applied. Here, therefore, in stratified-charge operation with high levels of excess air, injection takes place once in the compression stroke (similar to the "stratified-charge mode"), and then again in the combustion (power) cycle whereby the fuel injected here combusts very late and thus heats up the exhaust side and the exhaust system to a very high temperature.

A further important application is for heating up of the NO_x catalytic converter to temperatures in excess of 650 °C as needed to initiate the desulphurization of the catalytic converter. Here, it is imperative that dual injection is used since with conventional heating methods the high temperature which is required here cannot always be reached in all operating modes.

Ignition: An overview

The Otto-cycle engine is a gasoline internal-combustion engine with spark ignition (SI). An ignition spark is used to ignite the compressed A/F mixture in the combustion chamber and thus initiate its combustion. The ignition spark is in the form of a spark discharge between the spark-plug electrodes which extend into the combustion chamber. The ignition system is not only responsible for generating the high voltage needed for this spark discharge, but also for triggering the ignition spark at exactly the right instant in time.

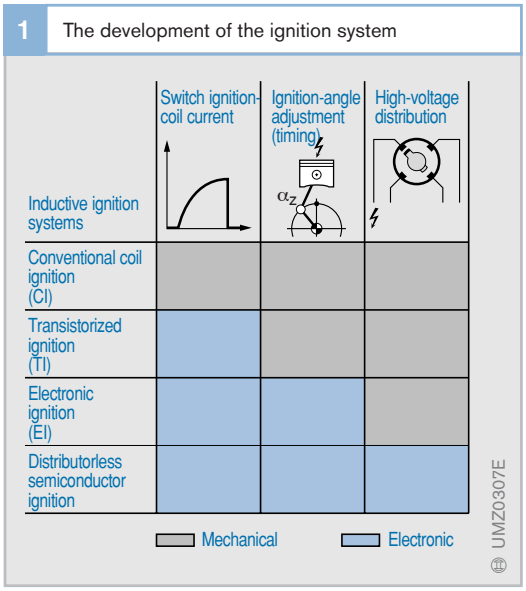
Survey

The most important characteristic values for the ignition of the A/F mixture are:

- Ignition angle, and
- Ignition energy.

The ignition angle is referred to crankshaft TDC. It defines the ignition point and therefore the inflammation or burning of the A/F mixture. It also has considerable influence on the gasoline engine's output power and its exhaust-gas emissions.

1 The development of the ignition system



A given voltage across the spark-plug electrodes, the ignition voltage, must be exceeded in order to generate the ignition spark in the combustion chamber. Depending on the engine's operating point and the condition of the spark plug, voltages as high as 30,000 V (turbocharged engine) are needed. Following the spark discharge, the spark energy is transferred to the A/F mixture and initiates the combustion process.

Inductive (coil) ignition systems have come to the forefront in passenger-car applications. In this form of ignition, the ignition energy is temporarily stored in the ignition coil's magnetic field and after having been transformed to a high enough voltage it is transferred to the A/F mixture at the ignition point.

Ignition systems with capacitive high-power energy storage are available for use with racing and high-performance engines. These store the ignition energy in the magnetic field of a capacitor.

Ignition systems: Development

Since they first came onto the market, there has been no letup in ignition-system development. This was, and is, the result of the ever-increasing demands made for higher engine outputs and improved exhaust-gas emissions. Here, electronics is continuing to play a more and more important role (Fig. 1).

Robert Bosch GmbH

Conventional coil ignition (CI)
(1934 ... 1986)
Mechanical breaker points in the ignition distributor control the flow of current through the ignition coil (charge coil and ignition). A mechanical (flyweight) advance mechanism and a vacuum unit define the ignition angle as a function of engine speed and load (mechanical ignition timing).

A mechanical rotor which rotates inside the ignition distributor is responsible for distributing the high voltage to the spark plugs (rotating high-voltage distribution).

Transistorized ignition (TI)
(1965 ... 1993)
The mechanical breaker points were replaced here by a non-wearing power transistor mounted in a transistorized trigger box. The transistor is triggered by an inductive or Hall sensor. The use of a transistor for switching means that the disadvantages due to wear at the mechanical breaker points are avoided.

Electronic ignition (EI)
(1983 ... 1998)
Although high-voltage distribution is still mechanical, the mechanical ignition timing is dispensed with. Engine speed and load are measured electronically and used as the input variables for an ignition map stored in a semiconductor memory. An ignition ECU with microcontroller is needed for triggering and control.

Distributorless semiconductor ignition
(1983 ... 1998)
With this ignition system, voltage distribution is no longer mechanical, but is performed electronically by the ignition ECU (static voltage distribution). This means that the ignition system no longer contains any components which are subject to wear.

As from 1998, all newly designed engines have been equipped with an engine ECU which combines distributorless semiconductor ignition and gasoline injection (Motronic, Fig. 2).

2 Section through a 4-cylinder engine with gasoline direct injection and distributorless semiconductor ignition

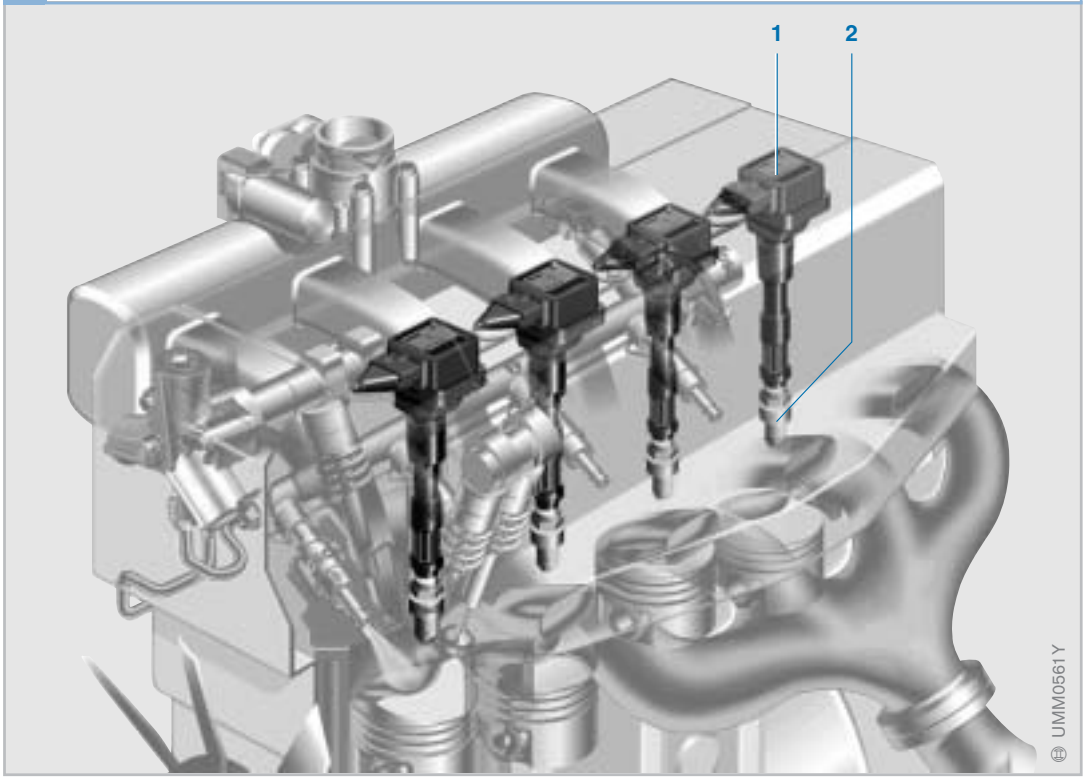


Figure 2
1 Spark-plug ignition coil
2 Spark plug

Robert Bosch GmbH

Coil ignition

The gasoline engine's (inductive) coil ignition system is responsible for the spark discharge at the spark-plug electrodes, and for the provision of enough energy for a powerful spark.

Survey

The ignition circuit of the coil ignition comprises the following components:

- Ignition driver stage (Fig. 1, Pos. 1),
- Ignition coil (2),
- High-voltage distributor,
- Spark plug (4), and
- Connecting devices and interference suppressors.

Modern ignition systems with static voltage distribution are no longer equipped with high-voltage distributors.

Using a coil ignition system with static voltage distribution and double-ended ignition coil as an example, Fig. 1 shows the principle design of the ignition circuit.

1 Using a coil ignition system with static voltage distribution and double-ended ignition coils as an example, Fig. 1 shows the basic design of the ignition circuit

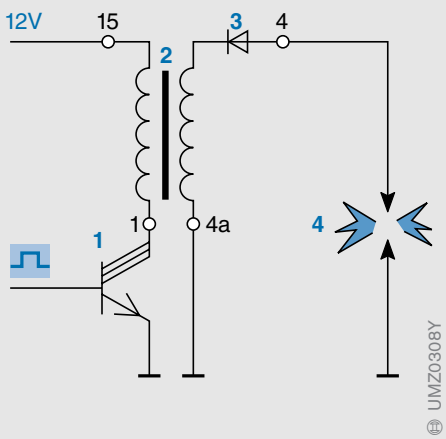


Figure 1
1 Ignition driver stage
2 Ignition coil
3 EFU diode (EFU = Switch-on spark suppression)
4 Spark plug

15,1,4,4a terminal designations
⎍ Triggering for the ignition driver stage

Ignition driver stage

Assignment

It is the job of the ignition driver stage to switch the ignition-coil current.

Design and operating concept

These driver stages are mostly in the form of a 3-stage power transistor. The "primary-current limitation" and "primary-voltage limitation" functions are integrated monolithically in the driver stage and serve to protect the ignition components against overload.

During operation, driver stage and ignition coil both heat up. In order not to exceed the permissible operating temperatures, it is necessary that appropriate measures are taken to ensure that the power loss is reliably dissipated to the surroundings even when outside temperatures are high. Primary-current limitation is only needed for limitation of current in case of fault (e.g. short circuit).

Internal and external ignition-driver stages are available. The former are integrated on the engine ECU printed-circuit board, and the latter are located in their own housing outside the engine ECU. Due to the costs involved, external driver stages are no longer used on new developments.

In addition, it is becoming increasingly common for the driver stages to be incorporated in the ignition coil.

Robert Bosch GmbH

Ignition coil

Assignment

The ignition coil stores the required ignition energy and generates the high voltage for the spark flashover at the ignition point.

Design

Today's state-of-the-art ignition coils are comprised of two magnetically-coupled copper windings (primary and secondary windings), an iron core assembled from sheet-metal laminations, and a plastic case. Depending upon design, the core can be of either the closed type (compact coil), or of the rod type (rod-type coil). The arrangement and location of the primary and secondary windings depends upon the coil's shape. In order to increase the insulation resistance, the secondary winding can be designed as a disc or chamber winding.

So as to ensure efficient insulation between primary and secondary winding, and between the windings and the case, the case is filled with epoxy resin. The design and construction of the ignition coil are adapted to the application in question.

Operating concept

The ignition coil functions according to Faraday's Law. The energy stored in the primary winding's magnetic field is transferred to the secondary winding by magnetic induction. Depending upon the turns ratio, voltage and current are transferred from the primary to the secondary winding (Fig. 2).

On the single-ended ignition coils for systems with rotating high-voltage distribution, one of the primary-winding terminals is connected to one of the secondary-winding terminals and then to Terminal 15 of the driving switch (economy connection). The other end of the secondary winding is connected to the ignition driver stage (Terminal 1). The secondary winding's other connection goes to the ignition distributor (Terminal 4). On the double-ended and dual-spark ignition coils used on ignition systems with static voltage distribution, primary and secondary windings are not connected. On the double-ended ignition coil, one end of the secondary winding (Term. 4a) is connected to ground, while the other end is directly connected to the spark plug. On the dual-spark ignition coil, each secondary-winding terminal is connected to a spark plug.

High-voltage generation

On modern ignition systems, the engine-management ECU switches on the ignition driver stage for the calculated dwell period, during which the coil's primary current increases to its desired value and in the process generates a magnetic field.

The magnitude of the primary current, together with the ignition coil's primary inductance, are decisive for the energy stored in this magnetic field.

2 Ignition coils: Schematic representations

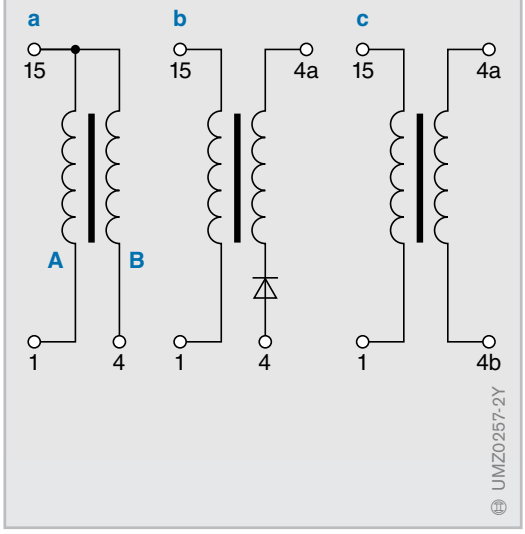


Figure 2
For rotating high-voltage distribution:
a Single-ended ignition coil

For static high-voltage distribution
b Double-ended ignition coil
c Dual-spark ignition coil

A Primary winding
B Secondary winding

At the moment of ignition (ignition point) the ignition driver stage interrupts the current flow. The resulting change in magnetic field induces the secondary voltage in the coil's secondary winding. The maximum possible secondary voltage is a function of the energy stored in the ignition coil, the winding capacitance, the coil's turns ratio, the secondary load (spark plug), and the primary-voltage limitation of the ignition driver stage.

The secondary voltage must in any case exceed the voltage level required for the flashover between the spark-plug electrodes (required ignition voltage). There must be adequate spark energy available to ignite the A/F mixture even when follow-up sparks are generated. These occur when the ignition spark is diverted by mixture turbulence and "breaks off" as a result.

When the primary current is switched on, an undesirable voltage of approx. 1...2 kV is induced in the secondary winding (switch-on voltage). This is of opposite polarity to the high voltage. Spark discharge at the spark plug (switch-on spark) must be avoided at all costs at this point.

On systems with rotating high-voltage distribution, the switch-on spark is effectively suppressed by the upstream distributor-rotor spark gap. In the case of static voltage distribution with double-ended ignition coils, a diode (EFU diode, Fig. 2b) in the high-voltage circuit stops the switch-on spark. With static voltage distribution, when dual-spark ignition coils are used, the switch-on spark is effectively suppressd by the high flashover voltage required for the series connection of two spark plugs. Additional measures need not be taken.

When the primary current is switched off, a 200...400 V self-induced voltage is generated in the secondary winding.

High-voltage distribution

Assignment

At the ignition point, the high voltage induced in the ignition coil must be available across the electrodes of the correct spark plug. This is the responsibility of the high-voltage distribution.

Rotating high-voltage distribution

In this form of high-voltage distribution, the voltage generated by a single ignition coil (Fig. 3a, Pos. 2) is mechanically distributed to the individual spark plugs (5) by an ignition distributor (3).

This form of distribution no longer has any significance for modern engine-management systems.

3 Principle of high-voltage distribution

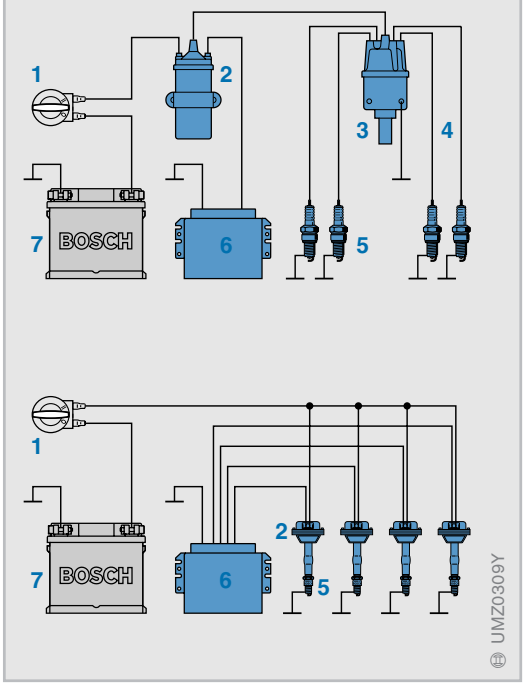


Figure 3

a Rotating high-voltage distribution
b Static high-voltage distribution with double-ended ignition coils
1 Ignition lock
2 Ignition coil
3 Ignition distributor
4 Ignition cable
5 Spark plug
6 ECU
7 Battery

Static voltage distribution

Mechanical components are dispensed with on distributorless (electronic or static) high-voltage distribution (Fig. 3b). The ignition coils are connected directly to the spark plugs and voltage distribution takes place at the ignition-coil primary side. This permits wear-free and loss-free voltage distribution. There are two versions of this form of voltage distribution.

Installations with double-ended ignition coils

Each cylinder is allocated its own spark-plug ignition coil and ignition driver stage. The engine ECU triggers the driver stage in accordance with the firing sequence.

Since there are no distributor losses, these ignition coils can be designed to be very small. Preferably, they are mounted directly above the spark plug.

The static voltage distribution with double-ended ignition coils can be applied universally irrespective of the number of engine cylinders. There are no limitations on the ignition-timing adjustment range, although this system must also be synchronized to the camshaft by means of a camshaft sensor.

Installations with dual-spark ignition coils

One ignition driver stage and one coil are allocated to two cylinders. The ends of the secondary winding are each connected to a spark plug in different cylinders. The cylinders have been chosen so that when one cylinder is in the compression stroke the other is in the exhaust stroke (applies only for engines with an even number of cylinders). Spark discharge takes place at each spark plug at the moment of ignition (ignition point). Care must be taken that the spark which takes place during the exhaust stroke does not ignite residual gas or fresh gas which has just been drawn in. Although this precautionary measure leads to a limitation in the the ignition-timing adjustment range, it is not necessary to synchronize the system to the camshaft.

Spark plugs

Assignment

The spark plug generates a spark which ignites the A/F mixture in the combustion chamber.

Design and operating concept

The spark plug (Fig. 4) is a ceramic-insulated, gastight high-voltage lead-through into the combustion chamber. It is provided with a ground electrode (2) and a center electrode (1).

The type of spark is determined by the position of the ground electrode(s). If this is opposite to the center electrode one speaks of an air-gap spark plug (a). When the ground electrode(s) is/are located to the side of the center electrode, this results in a side-electrode air-gap spark plug (b) or in the surface air-gap spark plug (c) or the purely surface-gap spark plug (d).

4 Spark plug (partial section) and spark gap

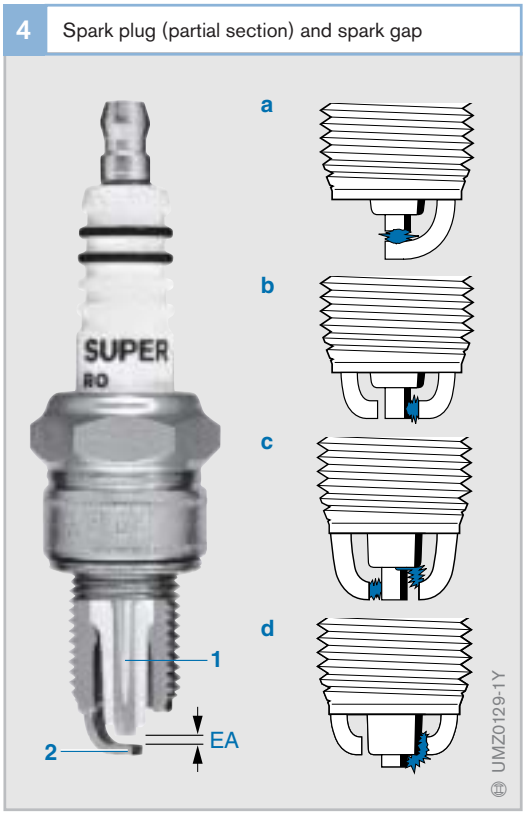


Figure 4
1 Center electrode
2 Ground electrode
a Air spark gap with front electrode
b Air spark gap with side electrode
c Surface air-gap (air spark or surface spark possible)
d Surface spark gap
EA Spark gap

Robert Bosch GmbH

After interrupting the primary current at the moment of ignition (ignition point), the voltage in the ignition coil's secondary winding increases very rapidly (approx. 30 µs, Fig. 5) to the ignition voltage. As soon as the required ignition voltage is exceeded, the spark gap between center and ground electrode becomes conductive. The capacitances in the secondary circuit which have charged up to ignition voltage (spark plug, ignition cable, and ignition coil) discharge abruptly in the form of a spark across the electrodes. Within a typical spark duration of 1...2 ms, the energy stored in the ignition coil is converted in a glow discharge (spark tail). The residual energy in the ignition coil then decays completely in a post-oscillation phase.

Spark-plug wear

During normal engine operation, the spark-plug electrodes are subject to wear as a result of the erosion stemming from the spark current and corrosion due to the hot gases in the combustion chamber. This wear enlarges the spark gap and the required ignition voltage increases as a result. Independent of the operating mode, up until the end of the prescribed spark-plug replacement interval, there must always be adequate secondary voltage available from the ignition system to reliably provide for this ignition voltage.

5 Voltage curve at the spark-plug electrodes

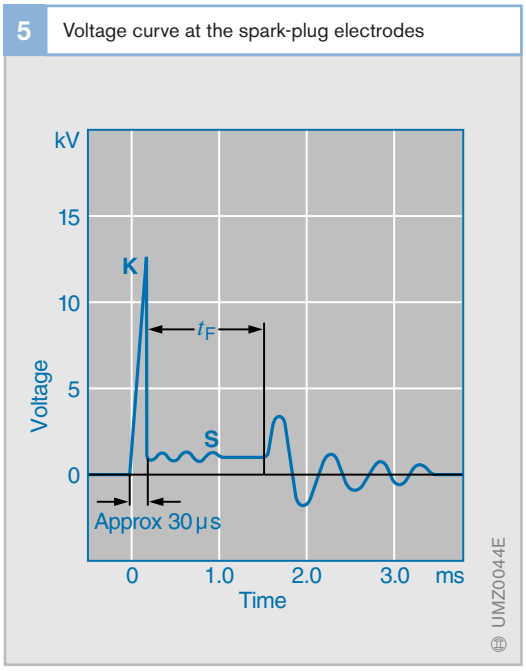


Figure 5
K Spark head
S Spark tail
t_F Spark duration

Electrical connection and interference-suppressor devices

Ignition cable

The high voltage generated in the ignition coil must be delivered to the spark plug. Special, plastic-insulated, high-voltage-proof cables with special plugs for contacting the high-voltage components, are used with ignition coils which are not mounted directly on the spark plug.

Since, for the ignition system, each high-voltage line represents a capacitive load which reduces the available secondary voltage, the ignition cables must be kept as short as possible.

Interference suppressors, shielding

The pulse-shaped discharge which occurs at every spark flashover at spark plug or ignition distributor (in the case of rotating high-voltage distribution) is a source of interference. Interference suppression resistors in the high-voltage circuit limit the discharge peak current. In order to minimise the interference radiation from the high-voltage circuit, the suppression resistors should be installed as close as possible to the interference source.

Normally, the interference resistors are integrated in the spark-plug connectors, in the plugs at the other end of the ignition cable and, when high-voltage distribution is used, in the distributor rotor. Spark plugs are also available which feature an integral suppression resistor. Increasing the secondary-circuit resistance, though, leads to increased energy losses in the ignition circuit and therefore to lower levels of spark energy at the spark plug.

Interference radiation can be even further reduced by partially or completely screening the ignition system.

Robert Bosch GmbH

Ignition voltage

This is the level of voltage across the spark-plug electrodes required to cause spark discharge between them. It depends upon a number of factors:

- Density of the A/F mixture in the combustion chamber, and therefore also the ignition point,
- Composition of the A/F mixture (excess-air factor, Lambda value),
- Flow velocity and turbulence,
- Electrode geometry,
- Electrode material,
- Electrode gap.

Care must be taken that the ignition system provides the required ignition voltage irrespective of operating conditions.

Ignition energy

The breaking current and the ignition-coil parameters define the energy stored by the ignition coil and then made available as ignition energy in the ignition spark. The ignition energy has a decisive influence upon the ignition of the mixture. Good A/F-mixture ignition is the prerequisite for high-performance engine operation coupled with low levels of toxic emissions. These requirements place high demands on the ignition system.

Energy balance of a single ignition process

The energy stored in the ignition coil is released as soon as the ignition spark is initiated. This energy is divided into two different sections.

Spark head

The energy E which is stored in the ignition circuit's secondary-side capacity C, is released abruptly at the ignition point, and increases as the square of the applied voltage U ($E = 1/2\ CU^2$). Fig. 6 therefore shows a square-law curve.

6 Energy balance of an ignition process without shunt, resistance and Zener losses

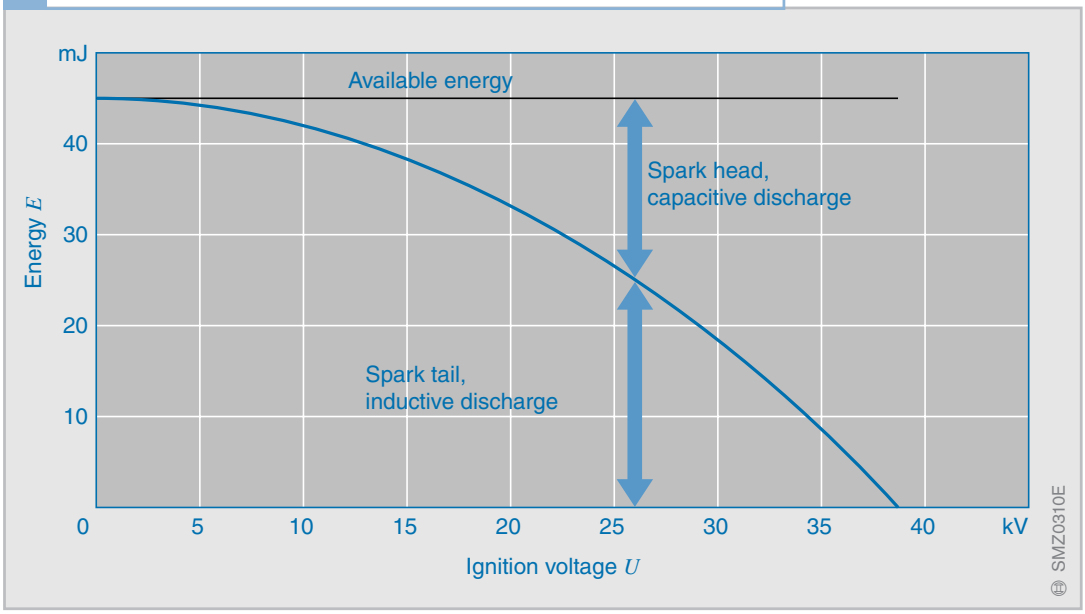


Figure 6
The energy values apply for an imaginary ignition system with an ignition-coil capacity of 35 pF, an external load of 25 pF, and a secondary inductance of 15 H.

Robert Bosch GmbH

Spark tail

The rest of the energy stored in the ignition coil (inductive share) is then released. This energy is the difference between the total energy stored in the ignition coil, and the energy released by capacitive discharge.

This means that the higher the required ignition voltage, the larger is the proportion of total energy in the spark head.

In certain cases, when the required ignition voltage is very high due for instance to badly worn spark plugs, the energy stored in the spark tail no longer suffices to completely burn an already ignited A/F mixture or, by means of follow-up sparks, re-ignite a flame that has been extinguished.

Further increases in the required voltage lead to the misfire limit being reached. The available energy in the spark head no longer suffices to generate a spark discharge and decays away as a damped oscillation (ignition misfire).

Shunt losses

Fig. 6 on the previous page shows a simplified representation of the existing conditions. The suppression resistors themselves, and the ohmic resistances in the ignition coil and ignition lines, cause losses which are then not available as ignition energy.

Further losses result from shunt resistances which can be caused by contamination at the high-voltage connections, as well as by deposits and soot on the parts of the spark plug projecting into the combustion chamber.

The severity of the shunt losses depends upon the required ignition voltage. The higher the voltage applied to the spark plug, the higher are the currents which are lost through shunt resistances.

Igniting the A/F mixture

Under ideal conditions, provided that the A/F mixture is stationary, homogeneous and stoichiometric, for each individual ignition process an energy of approx. 0.2 mJ is required to ignite the mixture by means of electric spark. Under such conditions, rich or lean mixtures need more than 3 mJ.

The energy that is actually required to ignite the A/F mixture (the ignition energy) is only a fraction of the total energy in the ignition spark. On conventional ignition systems, when high break-down voltages are concerned, energies in excess of 15 mJ are needed to generate the high-voltage spark discharge at the ignition point. Further energy is required to compensate for losses, due for instance to contamination shunts at the spark plugs, and in order to maintain the spark for a given period of time. These requirements amount to ignition energies of at least 30...50 mJ, a figure which corresponds to an energy level of 60...120 mJ stored in the ignition coil.

A/F-mixture turbulences such as occur in the stratified-charge mode with gasoline direct injection, can divert the ignition spark to such an extent that it extinguishes. A number of follow-up sparks are then needed to ignite the A/F mixture, and this energy must also be provided by the ignition coil.

The more air there is in a lean A/F mixture, the more difficult it is to ignite it. This fact leads to a particularly high level of energy being needed on the one hand to cover the higher ignition-voltage requirements, and on the other to ensure that spark duration is as long as necessary.

If insufficient energy is available, the A/F mixture does not ignite and this leads to combustion misses.

Robert Bosch GmbH

These facts mean that enough ignition energy must be made available so that the A/F mixture ignites reliably even under the most adverse conditions. In such cases, igniting a small A/F-mixture cloud in the vicinity of the spark plug can suffice to initiate ignition and combustion of the rest of the A/F mixture in the cylinder.

Influences on the ignition characteristic

Efficient mixture formation and ease of access to the ignition spark improve the ignition characteristic, as do an extended spark duration, a long spark, and a wide electrode gap. Mixture turbulence can also be an advantage provided enough energy is available for follow-up ignition sparks should these be needed. Turbulence supports rapid flame-front distribution in the combustion chamber, and with it the rapid and complete combustion of all the A/F mixture.

Spark-plug contamination is also of considerable importance. If the spark plugs are very dirty, energy flows from the ignition coil and through the spark-plug shunt (deposits) during the time in which the high voltage is being built up. This reduces the high voltage, shortens the spark duration, and has a negative effect upon the exhaust gas. In extreme cases, this can lead to ignition misfire if the spark plugs are badly contaminated or wet.

Ignition misfire leads to combustion miss which increases both fuel consumption and exhaust-gas emissions. The catalytic converter can also be damaged.

Ignition point

About two milliseconds elapse between the moment the ignition spark is generated and complete combustion. These figures apply as long as the A/F mixture composition remains unchanged. Therefore, along with increasing engine speed, ignition must also take place at an earlier and earlier point referred to the crankshaft angle.
Poor cylinder charge means that the A/F mixture's ignition characteristic deteriorates accordingly. This leads to increased ignition lag so that the ignition point has to be advanced even further. For the best-possible torque output, the ignition angle must be chosen so that main combustion, and with it the peak pressure, takes place after Top Dead Center (TDC), whereby care should be taken that the engine does not knock (Fig. 7).

In the stratified-charge mode (gasoline direct injection), the range for the variation of the ignition point is limited due to the end of injection and the time needed for A/F-mixture formation during the compresion stroke.

7 Pressure curve in the combustion chamber for different ignition angles (ignition points)

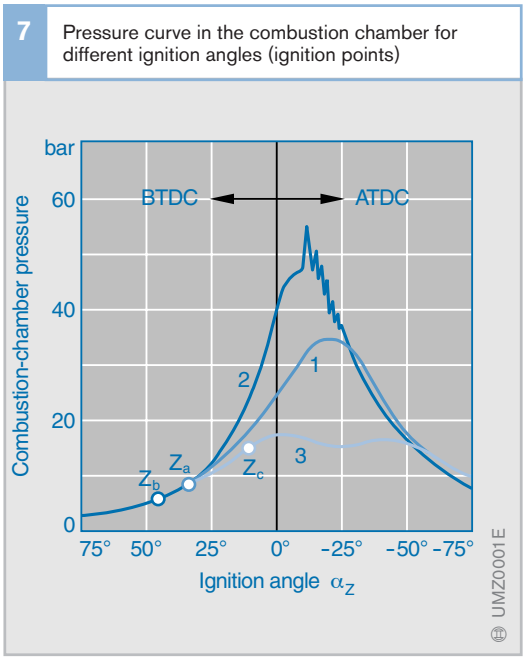


Figure 7
1 Ignition Z_a at the right moment in time
2 Ignition Z_b too early (combustion knock)
3 Ignition Z_c too late

Catalytic emissions control

Emission-control legislation defines the limits for the toxic agents generated during the combustion process in the spark-ignition engine. Catalytic treatment of the exhaust gas is necessary in order to comply with these limits.

Overview

Before leaving the exhaust pipe, the exhaust gas flows through the catalytic converter installed in the exhaust-gas tract (Fig. 1, Pos. 3). Inside the converter, special coatings ensure that the toxic agents in the exhaust gas are chemically converted to harmless substances. Lambda oxygen sensors (2, 4) are used to measure the residual-oxygen content in the exhaust gas. These measured values are then applied in adjusting the A/F mixture so that the catalytic converter can work at maximum efficiency.

A number of different catalytic-converter concepts were applied in the past years. The three-way catalytic converter represents the state-of-the-art for engines with homogeneous A/F mixture distribution and operation at $\lambda = 1$. Engines which run with a lean A/F mixture also require a NO_x accumulator-type catalytic converter.

Oxidation-type catalytic converter

In this type of catalytic converter, the hydrocarbons and the carbon monoxide in the exhaust gas are converted by oxidation (burning) into water vapor and carbon dioxide. The oxygen needed for the burning process is already present in the case of a lean A/F mixture ($\lambda > 1$) or by blowing air into the exhaust-gas tract upstream of the converter. The oxidation converter cannot convert the oxides of nitrogen (NO_x).

Oxidation-type catalytic converters were first introduced in 1975 in order to comply with the exhaust-gas legislation in force in the USA at that time. Today, catalytic converters which operate exclusively with oxidation principles are used only very rarely.

1 Exhaust-gas tract with Lambda oxygen sensors and a three-way catalytic converter installed in the immediate vicinity of the engine

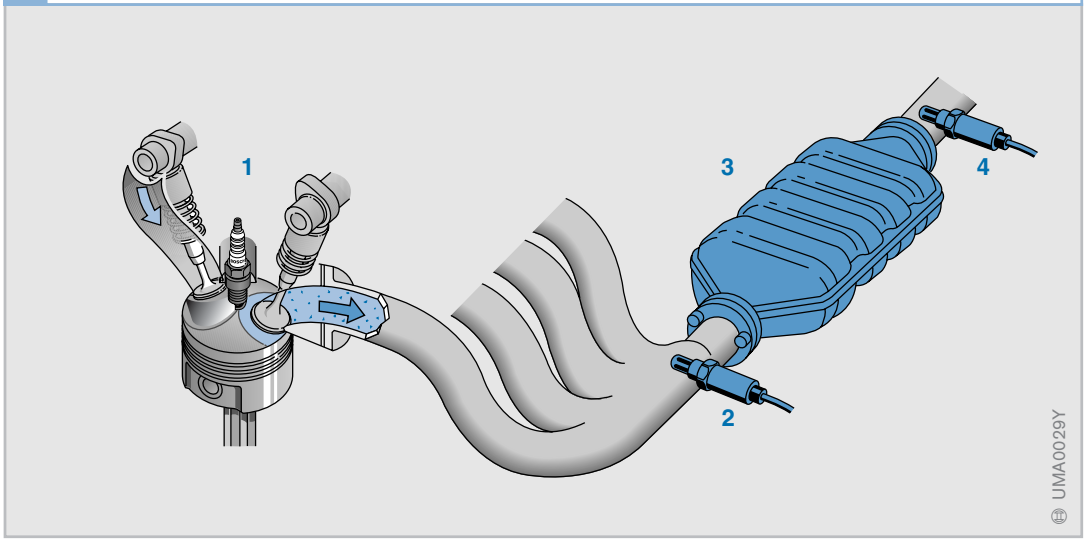


Figure 1
1 Engine
2 Lambda oxygen sensor upstream of the catalytic converter (two-step sensor or broad-band sensor depending upon system)
3 Three-way catalytic converter
4 Two-step lambda oxygen sensor downstreaam of the catalytic converter (only on systems with lambda two-sensor control)

Robert Bosch GmbH

Three-way catalytic converter

The three-way catalytic converter is installed in the exhaust-emission control systems of manifold-injection engines and gasoline direct-injection engines.

Assignment

Three toxic components are generated during the combustion of the A/F mixture: HC (hydrocarbons), CO (carbon monoxide), and oxides of nitrogen (NO_x). It is the job of the three-way catalytic converter to convert these into harmless components. The products which result from this converion are H_2O (water vapor), CO_2 (carbon dioxide), and N_2 (nitrogen).

Operating concept

The toxic components are converted in two phases: Firstly, the carbon monoxide and the hydrocarbons are converted by oxidation (Fig. G, Equations 1 and 2). The oxygen needed for the oxidation process is available in the exhaust gas in the form of the residual oxygen resulting from incomplete combustion, or it is taken from the oxides of nitrogen whereby these reduce as a result (Fig. G, Equations 3 and 4).

The concentration of the toxic substances in the untreated exhaust gas is a function of the excess-air factor λ (Fig. 2a). For carbon monoxide and hydrocarbons (HC), the conversion level increases steadily along with increasing excess-air factor (Fig. 2b). At $\lambda = 1$, there is only a very low level of toxic components in the untreated exhaust gas. With high excess-air factors ($\lambda > 1$), the concentration of these toxic components remains at this low level.

Conversion of the oxides of nitrogen (NO_x) is good in the rich range ($\lambda < 1$) . The lowest levels of NO_x are present during stoichiometric operation ($\lambda = 1$). Even a small increase in the exhaust-gas oxygen content as caused by operation at $\lambda > 1$ impedes the nitrogen reduction and causes a sharp increase in its concentration.

In order to maintain the three-way catalytic converter's conversion level for all three toxic substances at as high a level as possible, these must be present in a chemical balance in the exhaust gas. This means that the A/F mixture composition must have a stoichiometric ratio of $\lambda = 1$, so that the "window" for the A/F mixture ratio l is necessarily very restricted. A/F mixture formation must be controlled by a Lambda closed-loop control circuit.

G Reaction equations in the three-way catalytic converter

(1)	$2\ CO$	$+\ O_2$	$\rightarrow 2\ CO_2$	
(2)	$2\ C_2H_6$	$+\ 7\ O_2$	$\rightarrow 4\ CO_2$	$+\ 6\ H_2O$
(3)	$2\ NO$	$+\ 2\ CO$	$\rightarrow N_2$	$+\ 2\ CO_2$
(4)	$2\ NO_2$	$+\ 2\ CO$	$\rightarrow N_2$	$+\ 2\ CO_2 + O_2$

2 Toxic components in the exhaust gas

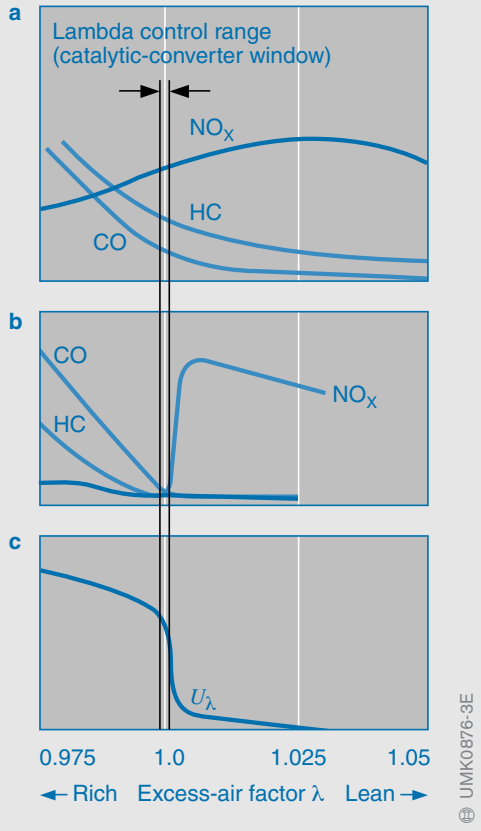


Figure 2
a Before catalytic aftertreatment (untreated exhaust gas)
b After catalytic aftertreatment
c Voltage characteristic of the two-step Lambda sensor

Robert Bosch GmbH

Design and construction

The catalytic converter (Fig. 3) comprises a steel casing (6), a substrate (5), and the active catalytic noble-metal coating (4).

Substrates

Two substrate systems have come to the forefront

Ceramic monoliths

These ceramic monoliths are ceramic bodies containing thousands of narrow passages through which the exhaust gas flows. The ceramic is a high-temperature-resistant magnesium-aluminum silicate. The monolith, which is highly sensitive to mechanical tension, is fastened inside a sheet-steel housing by means of mineral swell matting (2) which expands the first time it is heated up and firmly fixes the monolith in position. At the same time the matting also ensures a 100 % gas seal.

Ceramic monoliths are at present the most commonly used catalyst substrates.

Metallic monoliths

The metallic monolith (metal catalytic converter) is an alternative to the ceramic monolith. It is made of finely corrugated, 0.05 mm thin metal foil which is wound and soldered in a high-temperature process. Thanks to its thin walls, more passages can be accomodated inside the same area, which means less resistance to exhaust-gas flow, a fact which is important in the case of high-performance engines.

Coating

The ceramic and metallic monoliths require an aluminum oxide (Al_2O_3) substrate coating, the so-called "Washcoat" (4). This coating serves to increase the converter's effective surface area by a factor of around 7000. On the oxidation catalytic converter, the effective catalytic coating applied to the substrate contains the noble metals platinum and/or palladium. On the three-way converter, rhodium is also applied. Platinum and palladium accelerate the oxidation of the hydrocarbons (HC) and of the carbon monoxide. Rhodium accelerates the reduction of the oxides of nitrogen (NO_x).

Depending upon the engine's displacement, a catalytic converter contains about 1...3 g of noble metal.

3 Three-way catalytic converter with Lambda oxygen sensor

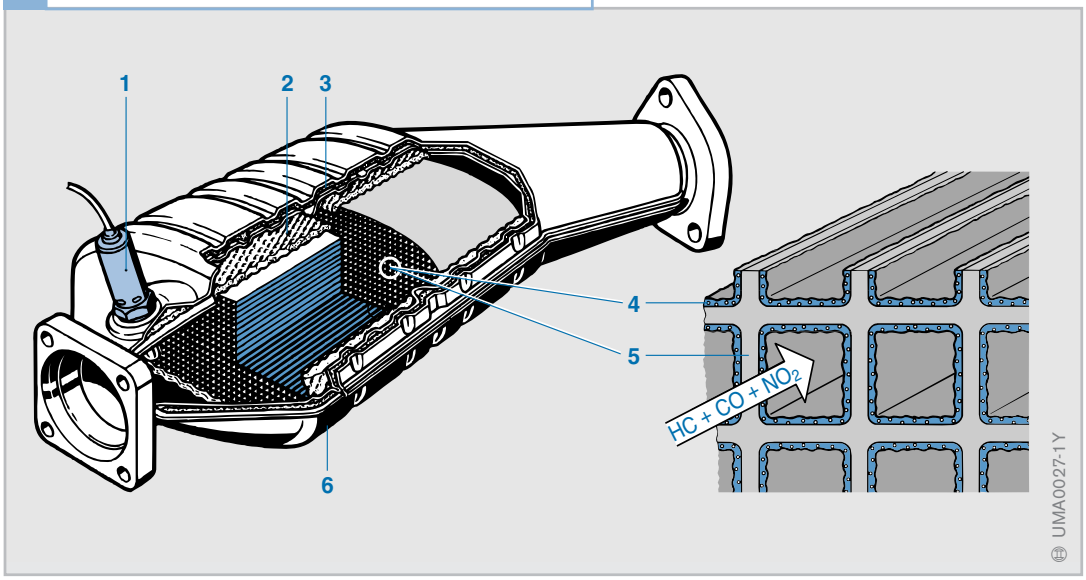


Figure 3

1 Lambda oxygen sensor
2 Swell matting
3 Thermally insulated double shell
4 Washcoat (Al_2O_3 substrate coating) with noble-metal coating
5 Substrate (monolith)
6 Housing

Operating conditions

Operating temperature

The catalytic converter's temperature plays a decisive role in emission-control efficiency. Considering a three-way catalytic converter, no worthwhile conversion of toxic substances takes place until temperature exceeds 300 °C. Operation within a temperature range of 400...800 °C is ideal with regard to high conversion levels and a long service life.

At temperatures between 800...1000 °C, thermal aging is accelerated due to the sintering of the noble metals and of the Al_2O_3 substrate layer, and this leads to a reduction of the effective surface. The time spent at 800...1000 °C is of vital importance, and above 1000 °C thermal aging increases drastically and leads to the catalytic converter becoming practically 100 % ineffective.

Engine malfunction (ignition misfire) can cause the temperature inside the catalytic converter to exceed 1400 °C. Since such temperatures melt the substrate and completely destroy the catalyst, it is imperative that the ignition system is highly reliable and maintenance-free. Modern engine-management systems are able to detect ignition and combustion miss, and in such cases interrupt the fuel injection to the cylinder concerned so that unburned A/F mixture cannot enter the exhaust-gas tract.

Unleaded fuel

Another prerequisite for long-term operation is the use of unleaded fuel. Otherwise, lead compounds are deposited in the pores of the active surface and reduce their number. Residues from the engine oil can also "poison" the catalyst and damage it so far that it becomes ineffective.

Installation point

Strict emissions-control legislation demands special concepts for heating the catalytic converter when the engine is started. The catalytic converter's installation point is determined by such concepts (for instance, secondary-air injection, shift of the timing in the "retard" direction). The three-way catalytic converter's sensitivity regarding operating temperature limits the choice of installation point. The temperature conditions needed for a high conversion level make it absolutely imperative that the three-way converter is installed close to the engine.

In the case of the three-way catalytic converter, a configuration featuring a "pre-cat" near the engine followed by a second (main) underfloor catalytic converter has come to the forefront. Catalytic converters near the engine demand that their coating techniques be optimized to provide for high-temperature stability. Underfloor converters on the other hand, require optimisation in the so-called "low light-off" direction (low start-up temperature) and good NO_x conversion characteristics.

An alternative is available with just one "overall" catalytic converter which is then installed close to the engine.

Effectiveness

For a spark-ignition engine with homogeneous mixture distribution operating at $\lambda = 1$, catalytic treatment of the exhaust gas using a three-way catalytic converter is at present the most effective emission-control method. Included in this system is the Lambda closed-loop control which monitors the composition of the A/F mixture. Using the three-way catalytic converter, the pollutant emissions of carbon monoxide, hydrocarbons, and oxides of nitrogen can be practically eliminated provided the engine operates with homogeneous A/F-mixture distribution and at stoichiometric A/F ratio. Notwithstanding the fact that it is not always possible to comply fully with these operating requirements, one can still presume an average pollutants reduction of more than 98 %.

Robert Bosch GmbH

NO_x accumulator-type catalytic converter

Assignment

During lean-burn operation, it is impossible for the three-way catalytic converter to completely convert all the oxides of nitrogen (NO_x) which have been generated during combustion. In such cases namely, the oxygen that is needed for the oxidation of the carbon monoxide and of the hydrocarbons is not split off from the oxides of nitrogen but instead is taken from the high level of residual oxygen in the exhaust gas. The NO_x accumulator catalytic converter reduces the oxides of nitrogen in a different manner.

Design and special coating

The NO_x accumulator-type catalytic converter is similar in design to the conventional three-way converter. In addition to the platinum and rhodium coatings, the NO_x converter is provided with special additives which are capable of accumulating oxides of nitrogen. Typical accumulator materials are the oxides of potassium, calcium, strontium, zirconium, lanthanum, and barium.
The coating for NO_x accumulation and for the 3-way catalytic converter can be applied on a common substrate.

Operating concept

At $\lambda = 1$, due to the noble-metal coating the NO_x converter operates the same as a three-way converter. In lean exhaust gases though it also converts the non-reduced oxides of nitrogen. This conversion is not a continuous process as it is with the hydrocarbons and the carbon monoxide, but instead takes place in three distinct phases:

1. NO_x accumulation (storage),
2. NO_x release, and
3. Conversion.

NO_x accumulation (storage)

On the surface of the platinum coating, the oxides of nitrogen (NO_x) are oxidized catalytically to form nitrogen dioxide (NO_2). The NO_2 then reacts with the special oxides on the catalyst surface and with oxygen (O_2) to form nitrates. For instance, NO_2 combines chemically with barium oxide (BaO) to form barium nitrate $(NO_3)_2$ (Fig. G, Equation 1). This enables the NO_x converter to accumulate the oxides of nitrogen which have been generated during engine operation with excess air.

There are two methods in use to determine when the NO_x converter is full and the accumulation phase has finished:

- Taking the catalyst temperature into account (Fig. 1, Pos. 4), the model-based method calculates the quantity of stored NO_x.
- An NO_x sensor (6) downstream of the NO_x converter continually measures the NO_x concentration in the exhaust gas.

NO_x removal and conversion

The more NO_x that is stored, the less the ability to chemically bind further nitrogens of oxide. This means that regeneration must take place as soon as a given level is exceeded, in other words the accumulated oxides of nitrogen must be released and converted. To this end, the engine is run briefly in the rich homogeneous mode ($\lambda < 0.8$). The processes for releasing the NO_x and converting it to nitrogen and carbon dioxide take place separately from each other. H_2, HC, and CO are used as reducing agents. Reduction is slowest with HC and most rapid with H_2. NO_x release takes place as follows, whereby the following description applies with carbon monoxide (CO) as the reducing agent: The carbon monoxide reduces the nitrate (e.g. barium nitrate $Ba(NO_3)_2$ to an oxide (e.g. barium oxide BaO). This leads to the generation of carbon dioxide (CO_2) and nitrogen monoxide (NO) (Fig. G, Equation 2).

Robert Bosch GmbH

G Reaction equations for the NO_x accumulation phase (1), removal phase (2), and conversion phase (3)

(1) $2\,BaO + 4\,NO_2 + O_2 \rightarrow 2\,Ba(NO_3)_2$

(2) $Ba(NO_3)_2 + 3\,CO \rightarrow 3\,CO_2 + BaO + 2\,NO$

(3) $2\,NO + 2\,CO \rightarrow N_2 + 2\,CO_2$

Subsequently, using the carbon monoxide (CO), the rhodium coating reduces the NO_x to nitrogen and carbon dioxide (CO_2) (Fig. G, Equation 3).

There are two different methods for determining the end of the NO_x-release phase:

- The model-based method calculates the quantity of NO_x still held by the converter.
- A Lambda oxygen sensor (Fig. 1, Pos. 6) downstream of the converter measures the exhaust-gas oxygen concentration and outputs a voltage jump from "lean" to "rich" when conversion has finished.

Operating temperature and installation point

The NO_x converter's ability to accumulate/store NO_x is highly dependent upon temperature. Accumulation reaches its maximum between 300 and 400 °C, which means that the favorable operating-temperature range is much lower than that of the three-way catalytic converter. For catalytic emissions control, therefore, two separate catalytic converters must be installed - a three-way pre-cat near the engine (Fig. 1, Pos. 3), and an NO_x accumulator-type main converter (5) remote from the engine (underfloor cat).

Sulphur in the NO_x accumulator-type catalytic converter

The sulphur in gasoline presents the accumulator-type catalytic converter with a problem. The sulphur contained in the exhaust gas reacts with the barium oxide (accumulator material) to form barium sulphate. The result is that, over time, the amount of accumulator material available for NO_x accumulation diminishes. Barium sulphate is extremely resistant to high temperatures, and for this reason is only degraded to a slight degree during NO_x regeneration. When sulphurized gasoline is used therefore, desulphurization must be carried at regular intervals. Here, selective measures are applied to heat the converter to between 600 and 650 °C. For instance, the engine can be run in the "stratified-charge/cat-heating mode". Rich ($\lambda = 0.95$) and lean ($\lambda = 1.05$)

1 Exhaust-gas system with three-way catalyic converter as pre-cat, and downstream NO_x accumulator-type converter and Lambda oxygen sensors

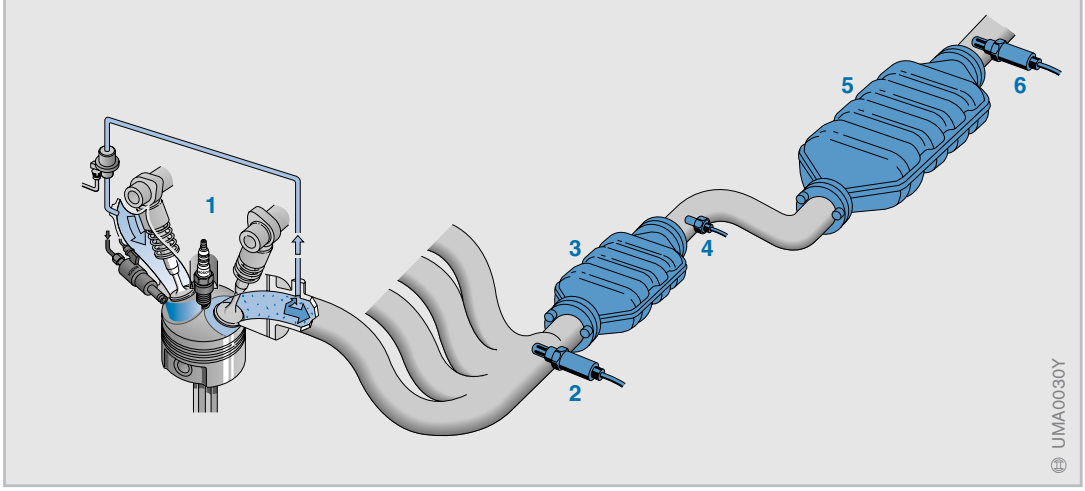


Figure 1
1 Engine with EGR system
2 Lambda oxygen sensor upstream of the catalytic converter
3 Three-way catalytic converter (pre-cat)
4 Temperature sensor
5 NO_x accumulator-type catalytic converter (main cat)
6 Two-step Lambda oxygen sensor, optionally available with integral NO_x sensor

Robert Bosch GmbH

exhaust gases are then passed through the cat one after the other. The barium sulphate reduces to barium oxide as a result.

Lambda control loop

Assignment

For systems which operate with only a single three-way catalytic converter, the pollutants must be in a state of chemical balance in order that the conversion level for all three pollutant constituents is as high as possible. This necessitates a stoichiometric A/F-mixture composition with $\lambda = 1.0$, which means that the "window" in which the A/F ratio must be located is very narrow. The only solution is to apply closed-loop control to the adjustment of the A/F mixture ratio. Open-loop control of fuel metering is not accurate enough.

Direct-injection gasoline engines are run with A/F mixtures which deviate from stoichiometric. Closed-loop control can also be used on these systems.

Design and construction

A Lambda oxygen sensor (Fig. 1, Pos. 3a) is located upstream of the pre-cat (4). The sensor signal U_{Sa} is inputted to the engine ECU (7). In order to do so, either a two-step Lambda sensor (two-step control) or a broad-band Lambda sensor (continuous-action Lambda control) must be used. A further Lamda oxygen sensor (3b) can be situated downstream of the main catalytic converter (5). This is always a two-step sensor, and it delivers the sensor signal U_{Sb}. This form of control is known as two-sensor control.

Operating concept

Using the Lambda control loop, deviations from a specific A/F-ratio can be detected and corrected. The control principle is based on the measurement of the residual oxygen in the exhaust gas. This is a measure for the composition of the A/F mixture supplied to the engine (2).

Two-step control

The sensor voltage U_{Sa} generated by the two-step Lambda oxygen sensor upstream of the pre-cat (4) is high in the rich range ($\lambda < 1$) and low in the lean range ($\lambda > 1$). Since the sensor voltage jumps abruptly at $\lambda = 1$, the two-step Lambda oxygen ensor can only differentiate between rich and lean A/F mixtures.

1 Functional diagram of the Lambda closed-loop control

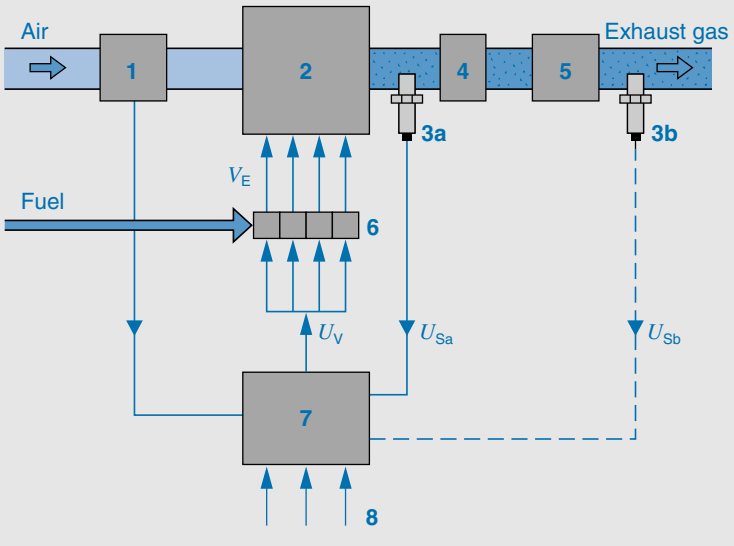


Figure 1
1 Air-mass meter
2 Engine
3a Lambda oxygen sensor upstream of the pre-cat (two-step Lambda sensor, or broad-band Lambda sensor)
3b Two-step Lambda sensor downstream of the main catalytic converter (only if required; on gasoline direct injection with integral NO_x sensor)
4 Pre-cat (three-way catalytic converter)
5 Main cat (On manifold injection: three-way converter; on gasoline direct injection: NO_x accumulator-type converter)
6 Injectors
7 Engine ECU
8 Input signals

U_S Sensor voltage
U_V Injector-triggering voltage
V_E Injected fuel quantity

The sensor output signal is converted to a binary signal in the engine ECU and used as the input signal for the Lambda closed-loop control as implemented using software. The Lambda control has a direct influence on the A/F mixture formation and sets the correct A/F ratio by adapting the injected fuel quantity. The manipulated variable comprises a step change and a ramp, and its control direction changes with each jump of the sensor voltage. In other words, a jump of the manipulated variable causes the A/F mixture to change. This change is first of all very abrupt, and then it follows a ramp. With a high sensor voltage ("rich" A/F mixture), the manipulated variable adjusts in the "lean" direction, and for a low sensor voltage ("lean" A/F mixture) in the "rich" direction. This so-called two-step control enables A/F mixture to be closed-loop controlled to values around $\lambda = 1$.

Shaping the manipulated variable's characteristic curve asymmetrically compensates for the Lambda sensor's typical false signal caused by variations in A/F mixture formation (rich/lean shift).

Continuous-action Lambda control

The broad-band Lambda sensor outputs a continuous voltage signal U_{Sa}. This means that not only the Lambda area (rich or lean) can be measured, but also the deviation from $\lambda = 1$ so that the Lambda control can react more quickly to an A/F mixture deviation. This leads to better control behaviour with highly improved dynamic response.

The broad-band Lambda oxygen sensor can measure A/F mixtures which deviate from $\lambda = 1$. This means that (in contrast to the two-step control), such A/F mixtures can also be controlled. The control range covers $\lambda = 0.7...3.0$ so that continuous Lambda control is suitable for the "rich" and "lean" operation of engines with gasoline direct injection.

Two-sensor control

When it is situated upstream of the pre-cat, the Lambda oxygen sensor (3a) is heavily stressed by high temperatures and untreated exhaust gas, and this leads to limitations in accuracy. On the other hand, locating the sensor downstream of the main catalytic converter (3b) means that these influences are considerably reduced.

The only problem here though is that a single downstream sensor would be far too "sluggish" due to the exhaust gases taking so long to reach it. The principle of two-sensor control relies upon the upstream sensor controlling the "lean" and "rich" shift, while the downstream sensor is part of a "slow" corrective closed control loop responsible for additive changes.

Lambda closed-loop control of gasoline direct injection

The NO_x accumulator-type catalytic converter has two different functions. During lean-burn operation, NO_x accumulation and CO oxidation must take place. In addition, at $\lambda = 1$, a stable three-way function is needed which provides for a minimum level of oxygen-accumulation.
The Lambda sensor upstream of the catalytic converter monitors the stoichiometric composition of the A/F mixture.
Together with the integrated NO_x sensor, the two-step Lambda sensor downstream of the NO_x accumulator converter not only takes part in the two-sensor control but also monitors the behaviour of the combination O_2 and NO_x accumulator (detection of the end of the NO_x release phase).

Robert Bosch GmbH

Catalytic-converter heating

Ignition timing towards “retard”

In order to keep the pollutant concentration in the exhaust gas down to a minimum, it is necessary that the catalytic converter reaches its operating temperature as soon as possible. One method is to adjust the ignition timing towards “retard”.

This step lowers the engine efficiency, and in doing so leads to hotter exhaust gases which then heat-up the converter.

Secondary-air injection

The unburnt components of the A/F mixture still present in the exhaust gas are burnt in the thermal afterburning process. With “lean” A/F mixtures, the oxygen required for this afterburning process is available in the exhaust gas in the form of residual oxygen. With “rich” A/F mixtures, as often needed for an engine which has not yet reached operating temperature, extra air (secondary air) is injected into the exhaust-gas passage to speed-up the catalytic-converter heating.

On the one hand, this exothermic reaction reduces the hydrocarbons and the carbon monoxide. On the other, afterburning also heats up the catalytic converter so that it quickly reaches its operating temperature. During the warm-up phase, this process considerably increases the conversion rate so that the catalytic converter is quickly ready for operation. Fig. 1 shows the curves of the hydrocarbon and carbon monoxide emissions in the first seconds of an emissions test, with and without secondary-air injection.

In line with present state-of-the-art, electric secondary-air pumps are used for secondary-air injection.

Post injection (POI)

On gasoline direct-injection engines, another method can be used for quickly bringing the catalytic converter up to temperature. In the “stratified-charge/cat-heating” operating mode, during stratified-charge operation with high levels of excess air a second injection of fuel takes place during the engine’s power cycle. This fuel is combusted late and causes considerable heat-up of the engine’s exhaust side and of the exhaust manifold. This means, that in those cases in which conventional measures (adjust ignition timing in the”retard” direction) do not suffice for complying with the stipulated exhaust-gas limits, the secondary-air pump used for manifold injection can be dispensed with.

1 Influence of secondary-air injection on CO and HC emissions

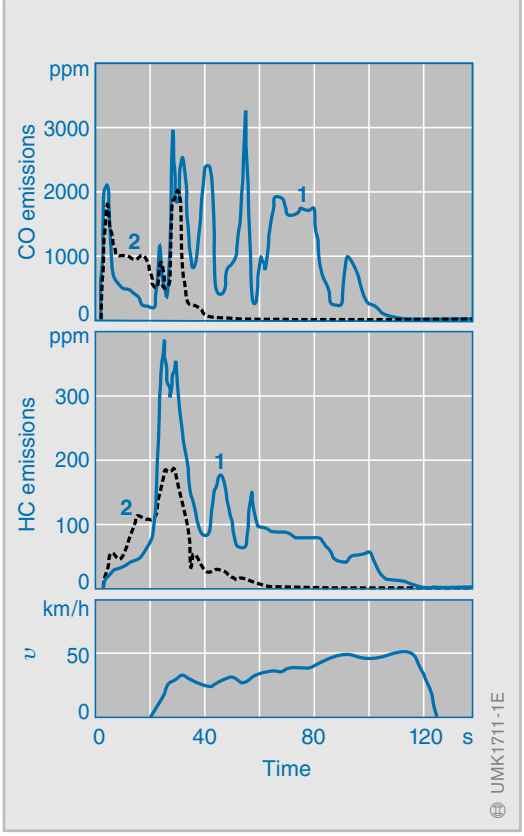


Figure 1
1 Without secondary-air injection
2 With secondary-air injection
n Vehicle speed

Robert Bosch GmbH

Index of technical terms

Technical terms

A
A/F mixture, 6
A/F ratio, 15
A/F-mixture cloud, 63
A/F-mixture distribution, 6
Air bypass actuator, 20
Air charge, 12
Air-mass meter, 49
Auto-ignition, 19

B
Boost pressure, 29
Bottom Dead Center (BDC), 4
Broad-band Lambda oxygen sensor, 83

C
Camshaft changeover, 23
Camshaft phase adjustment, 22
Canister-purge valve, 41
Carbon canister, 41
Carbon dioxide, 77
Carbon monoxide, 77
Catalytic converters, 76
Catalytic emissions control, 76
Center electrode (spark plug), 71
Centrifugal turbo-compressor, 29
Combustion knock, 19
Combustion process, 62
Common Rail, 55
Compression ratio, 6
Compression stroke, 4
Compressor, 29
Continuous-action Lambda control, 83
Conventional coil ignition (CI), 67
Conversion of toxic components, 77
Cylinder charge, 12
Cylinder (engine), 4
Cylinder-individual fuel injection (CIFI), 53

D
Delivery-quantity control valve, 56
Displacement-type compressor, 27
Distributorless semiconductor ignition, 65
Down-sizing, 33
Dual injection, 65
Dual spray, 52
Dual-spark ignition coil, 69
Dwell angle, 19
Dynamic supercharging, 26

E
Efficiency, 8
EGR valve, 25
Electric fuel pump, 36, 42
Electromagnetic fuel injectors, 50
Electronic ignition, 67
Electronic throttle control (EGAS), 21
Emission-control legislation, 76
Evaporative-emissions control system, 41
Excess-air factor (Lambda), 6
Exhaust stroke, 5
Exhaust valve, 5
Exhaust-gas recirculation (EGR), 25
Exhaust-gas turbine, 30
Exhaust-gas turbocharging, 30
External EGR, 25
Externally supplied ignition, 66

F
Follow-up spark, 74
Four-stroke principle, 4
Fresh gas, 12
Frictional losses, 9
Fuel consumption, 16, 25
Fuel filter, 36, 44
Fuel lines, 46
Fuel rail, 37, 45
Fuel supply, 36
Fuel tank, 36, 46
Fuel-pressure damper, 46
Fuel-pressure regulator, 36, 45
Fuel-supply system, 37

G
Gas-exchange valves, 4
Gasoline direct injection, 54
Ground electrode (spark plug), 71
Group fuel injection, 53

H
High-pressure injectors, 60
High-pressure pumps, 56
High-voltage distribution, 70
High-voltage generation, 69
Homogeneous mode, 64
Homogeneous/anti-knock mode, 65
Homogeneous and lean-burn mode, 65
Homogeneous and stratified-charge mode, 65
Hydrocarbons (HC), 77

I
Ignition angle, 18
Ignition cable, 72
Ignition coil, 69
Ignition distributor, 70
Ignition driver stage, 68
Ignition energy, 73
Ignition map, 18
Ignition point, 18, 70, 75
Ignition timing, 84
Ignition voltage, 72, 73
Induction stroke, 4
Inductive (coil) ignition system, 68
Inert gas, 13
Infinitely-variable valve timing, 24
Injection valves, 49
Injection-orifice plate, 50
Inner-gear pump, 42
Intake manifold, 26
Intake valve, 4, 49
In-tank unit, 40
Intercooling, 33
Interference-suppression resistor, 72
Internal EGR, 14, 23

K
Knock control, 19

L
Lambda closed-loop control, 83
Lambda control loop, 82
Lambda oxygen sensor, 76, 82
Lambda, 6
Lean-burn limit, 15
Low-pressure circuit, 39

M
Manifold chamber, 26
Manifold fuel injection, 48
Mechanical supercharging, 29
Monoliths (catalytic converter), 78
Multipoint fuel-injection systems, 34

N
Nitrogen, 77
Noble-metal coating, 78
Non-return valve, 36, 58
NO_x accumulator-type catalytic converter, 80
NO_x emissions, 25

Volkswagen Technical Site: http://vwts.ru http://vwts.info

O
Operating modes, 64
Output power, 7, 16
Overrun fuel cutoff, 17
Overrun, 17
Oxidation, 76
Oxides of nitrogen (NO_x), 76
Oxydation-type catalytic converter, 76

P
Palladium, 78
Pencil spray, 52
Peripheral pump, 43
Platinum, 78
Positive-displacement pump, 42
Post injection (POI), 84
Power (combustion) stroke, 4
Pre-cat, 79
Pressure-control valve, 55, 58
Presupply pump, 42
Primary pressure, 56
Primary winding (ignition coil), 69
Primary-current limitation, 68
Pumping losses, 9
p-V diagram, 8

R
Rail, 55, 56
Rail-pressure sensor, 59
Ram-tube supercharging, 26
Residual exhaust gas, 13
Rhodium, 78
Roller-cell pump, 42
Rotary-screw supercharger, 29
Rotating high-voltage distribution, 70

S
Secondary winding (ignition coil), 69
Secondary-air injection, 84
Sequential fuel injection, 53
Shunt losses, 74
Side-channel pump, 43
Simultaneous fuel injection, 53
Single-cylinder high-pressure pump, 57
Single-point injection (TBI), 35
Single-spark ignition coil, 69
Spark duration, 75
Spark head, 73
Spark length, 75
Spark plug, 71
Spark-plug ignition coil, 69
Spark tail, 74
Spiral-type supercharger, 29
Spray formation, 52
Spray offset angle, 52
Static voltage distribution, 71
Stoichiometric ratio, 15, 82
Stratified charge, 6
Stratified-charge mode, 64
Stratified-charge/cat-heating mode, 65
Sulphur charge, 81
Swirl air flow, 62
System pressure, 37

T
Tapered spray, 52
Thermal losses, 9
Three-cylinder high-pressure pump, 57
Three-way catalytic converter, 76, 77
Throttle device, 21
Throttle valve, 20
Throttling losses, 22
Top Dead Center (TDC), 4
Torque, 7
Trailing throttle, 17
Transistorized ignition (TI), 67
Tumble air flow, 62
Tuned-intake-tube charging, 27
Turbine pump, 43
Turbo flat spot, 33
Two-sensor control, 83
Two-step control, 82
Two-step Lambda oxygen sensor, 83
Types of injection, 53

U
Underfloor catalytic converter, 79

V
Valve overlap, 14
Valve timing, 5
Variable valve timing, 22
Variable intake-manifold geometry, 27
Volumetric efficiency, 14
VST supercharrger, 32
VTG supercharger, 31

W
Wall film, 17
Wall wetting, 17
Washcoat, 78
Wastegate supercharger, 31

Abbreviations

A
ATL: Exhaust-gas turbocharger

B
BDC: Bottom Dead Center
BPS: Boost-Pressure Sensor

C
CI: Coil Ignition
CIFI: Cylinder-Individual Fuel Injection
CO: Carbon monoxide
CO_2: Carbon dioxide

D
DI: Direct Injection
DR: Pressure regulator

E
ECU: Electronic Control Unit
EGAS: = ETC
EGR: Exhaust-Gas Recirculation
EI: Electronic Ignition
EKP: Electric fuel pump
ETC: Electronic Throttle Control
EI: Electronic Ignition

H
HC: Hydrocarbons
HDEV: High-pressure injector
HDP: High-pressure pump

I
IV: Intake Valve

L
LML: Lean Misfire Limit

M
MPI: Multi-Point Injection
MSV: Delivery-quantity control valve

N
NO_x: Oxides of nitrogen

P
POI: Post injection

R
PP: Peripheral Pump
RLFS: Returnless Fuel System
ROV: Rotating high-voltage distribution
RUV: Static voltage distribution
RZP: Roller-cell pump

S
SEFI: Sequential Fuel Injection
SI: Spark Ignition
SRE: Manifold fuel injection

T
TBI: Throttle-Body Injection
TDC: Top Dead Center
TI: Transistorized Ignition

V
VST: Variable Sleeve Turbine
VTG: Variable Turbine Geometry
VZ: Distributorless ignition

Z
ZP: Inner-gear pump